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OPTIMIZATION OF THE "KOOL-FIRE" HEAT PUMP

Report No. 87-63-K

D.F. Cuthbert
D.J. Young

ABSTRACT

This report is the Ontario Hydro published version of our Canadian Electrical Association report number 220 U 360. The original report abstract reads: "A new type of residential heating/cooling appliance was evaluated under laboratory conditions with a view to assessing its ultimate potential as a bivalent heating system. This novel device combines a heat pump with a gas burner (located under the outdoor coil) to form a combination heat-pump/condensing-gas furnace. Tests of the "as-delivered" machine's efficiencies and refrigerant-circuit thermodynamics suggested ways to improve the utilization efficiency and design of the unit. The heat pump was modified and retested to determine the effects of the changes. Several changes to the appliance are recommended and other considerations for the application of such a device are detailed."

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12-586-111	814.344	June 11, 1987	87-63-K

Ontario Hydro

Memorandum To SEE DISTRIBUTION
(of Report 87-63-K)

Date May 20, 1987

Location or Dept

File 814.344(p)

Subject Testing of the Kool-Fire Heat Pump

The attached Research Division Report, No. 87-63-K, contains a technical evaluation of an unusual Canadian "bivalent" heating system known as the Kool-Fire heat pump. This report provides detail on the operation of the device that may be of interest to your colleagues or staff. Overall, our assessment of this new technology has convinced us of the benefits of this type of heating load to Ontario's electricity customers and utilities alike. I wish to take the opportunity of this report's distribution to acquaint you with some of the further work now being carried out with respect to the Kool-Fire heat pump.

The Canadian Electrical Association has agreed to sponsor the manufacturer and us to implement selected improvements discovered in the course of the 87-63-K investigations. Some improvements, like extending the temperature range of heat pump operation so as to reduce the quantity of back-up fuel needed, have already been marketed. In fact, in its propane/electric version, this bivalent heating system may be one of the best air-source heat pumps (because it melts its own frost without needing a reverse cycle) and one of the best condensing furnace systems (because the propane is burned outside the house without risk of corrosion or explosion).

We are optimistic that the Kool-Fire system can be refined to the point where back-up propane consumption can be reduced so low that only one modest tank of fuel would be needed for a whole winter. By using electricity most of the time for heat pump heating or air-conditioning, and using propane only during the coldest weather, the resulting appliance should have an excellent electrical load characteristic with significant strategic conservation potential. To this end, we are already monitoring performance and load characteristics in a residential installation in Mississauga, and we plan to expand this monitoring to other areas of the Province. Both residential and commercial applications are possible.

We would be pleased to discuss the results of our tests with you at any time. In addition, we would appreciate receiving any comments you may wish to make on the report or any other facets of our work on this heating system.

A.F. Baljet
Manager
Electrical Research Department

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Attach

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To Dr. D. Mills
Director of Research

OPTIMIZATION OF THE "KOOL-FIRE" HEAT PUMP

D.F. Cuthbert, D.J. Young

The attached report is a record of the material submitted to the Canadian Electrical Association and published by them as CEA report 220 U 360. The work was carried out by Ontario Hydro Research Division between January 1984 and December 1985 and co-funded by the Canadian Electrical Association and Ontario Hydro.

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87-63

REPORT FOR

CANADIAN ELECTRICAL ASSOCIATION
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CEA No. 220 U 360

OPTIMIZATION OF THE "KOOL-FIRE" HEAT PUMP

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NOVEMBER 1986

ABSTRACT

A new type of residential heating/cooling appliance was evaluated under laboratory conditions with a view to assessing its ultimate potential as a bivalent heating system. This novel device combines a heat pump with a gas burner (located under the outdoor coil) to form a combination heat-pump/condensing-gas furnace. Tests of the "as-delivered" machine's efficiencies and refrigerant-circuit thermodynamics suggested ways to improve the utilization efficiency and design of the unit. The heat pump was modified and retested to determine the effects of the changes. Several changes to the appliance are recommended and other considerations for the application of such a device are detailed.

Keywords: Heat pump, residential heating system, bivalent heat pump, condensing-gas furnace.

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The authors greatly appreciate the co-operation and guidance of Murray Bond of B.C. Hydro and Walter Schimmel and Philip Critchlow of IREQ who were the technical advisors for the Canadian Electrical Association on this project.

The authors also wish to thank Mr. A. Galasso and Mr. S. Yap of Ontario Hydro for their contributions to this work.

EXECUTIVE SUMMARY

The performance of the latest design of the Kool-Fire 98 heating system (a combination heat pump and electrically-assisted condensing-gas furnace) was determined. The efficiency of this new "triple-split" machine was found to be significantly better than that of a previous model. In the heat pump mode at an outdoor temperature of 8.3°C (47°F), the steady-state coefficient of performance (COP) is 2.4 compared to 2.0 for the previous model. An even higher COP (2.8) was obtained when an optimal refrigerant charge was determined and when the suction filter was replaced by a liquid-line filter. It is estimated that the machine's COP could be increased further to around 3.1 by replacing the outdoor unit's blower with a more efficient fan and by lowering the refrigerant pressure drop through the outdoor coil. The improved machine's performance would rival that of some of the better commercially-available heat pumps. The unit's gas-burning-mode efficiency remained characteristically high at 95% and was relatively unaffected by outdoor temperature.

An extension to the project determined that a reduced charge condition did not adversely affect compressor-motor cooling. However, a somewhat higher charge was required for optimum cooling performance. Thus, for the configuration tested, the overall optimum charge would be 2.7 kg (6.0 lb) as opposed to the "as-delivered" charge of 4.2 kg (9.3 lb).

It was further shown that the overall seasonal efficiency of the Kool-Fire could be increased by using the air-source at lower outdoor temperatures. The current model operates as an air-source heat pump only above 8°C to avoid having to defrost the outdoor coil. However, a calculation showed that the air-source capacity of the improved Kool-Fire could deliver all the heating required for a typical house at much lower temperatures. Thus, providing it is possible to use the Kool-Fire's gas flame to defrost the outdoor coil, the Kool-Fire can be designed to make more use of efficient, air-source heating and use the gas flame only below the temperature where the house heating load exceeds the Kool-Fire's air-source capacity. In fact, partly as a result of the economic

evaluation presented in this and a previous study, the newest models of the Kool-Fire heat pump do operate in this manner.

Several characteristics distinguish the Kool-Fire heat pump from other types of heating systems. The advantages or disadvantages of each characteristic depend on the particular installation and application:

- A flue or chimney is not required and no indoor air is used for combustion since all the gas is burned outdoors.
- Complete replacement of an existing heating system with a Kool-Fire in a retrofit situation is made practicable because the single Kool-Fire indoor coil can do all the heating and cooling of the home. Conventional air-source, add-on heat pumps require a supplementary heat source in good condition (usually some type of furnace).
- Concerns about heat exchanger corrosion associated with condensing-type, fuel-burning appliances are mitigated with the Kool-Fire since the entire outdoor coil is at about 20°C during gas-burning mode and is continually washed with fresh condensate. Corrosion sites in ordinary condensing gas furnaces tend to be at locations in the heat exchanger which are at much higher temperatures (thereby increasing the chemical activity) and which are only intermittently exposed to the condensate (leading to higher acidity). Additional corrosion products in conventional condensing gas furnaces are reportedly associated with chloride contamination of the indoor combustion air. These are absent from the Kool-Fire because it inherently uses only outdoor air for combustion.
- Finally, since gas is burned during the coldest weather, the Kool-Fire consumes less than one quarter the power of a similarly-sized electric furnace or all-electric, air-source heat pump during the period of peak electric system demand on a winter-peaking utility.

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1 INTRODUCTION

The Kool-Fire model HC3 heat pump is a novel, residential heating device, manufactured in Canada, that acts as a hybrid heat pump and condensing-gas furnace. This bivalent heating system combines a conventional air-source heat pump with a gas burner that is located under the outdoor coil. The machine is sold as a complete home comfort system, providing all necessary heating in the winter and air-conditioning in the summer.

A previous version of this heat pump, model 32HB, was tested by Ontario Hydro Research Division and was described in OHRD Report 82-224-K (1). This original system was designed as a split-system heat pump with the compressor, reversing valve, and associated controls located in the outdoor cabinet along with the outdoor heat exchanger and gas burner. Model HC3, however, is a "triple-split" system with the compressor indoors in a separate cabinet. By locating the compressor indoors, the heat pump no longer requires an oil sump heater and losses from the compressor and discharge piping are minimized.

The unit operates as a heat pump using outdoor air as the heat source down to about 8°C. Below this temperature, a thermostat disables the outdoor fan and the gas burner is ignited, becoming the heat source for the outdoor coil. The compressor remains running to transfer the flame's heat indoors. Most of the higher heat of combustion of the gas can be recovered since the warm flue gases condense as they contact the relatively cool outdoor coil. This mode of operation eliminates the need for a defrost cycle. In addition, the full heating requirements of the home can be met by a single, indoor, heat-pump coil. Finally, since no gas is burned indoors, there is no requirement for a flue or chimney, or combustion-air supply.

As part of CEA's on-going program of heat pump research and development, Ontario Hydro's facilities were employed to carry out an assessment of

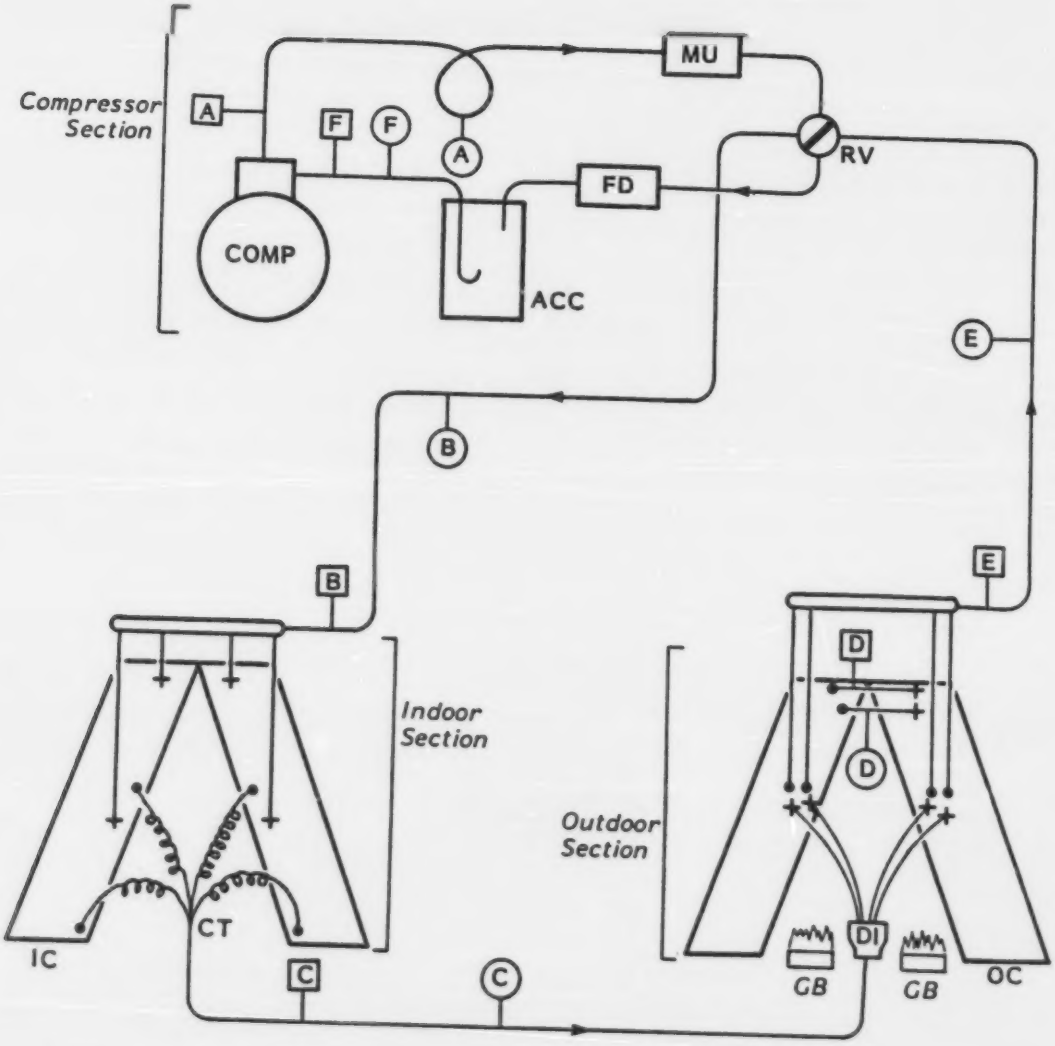
the performance of and potential improvements to this device. The following steps were carried out.

1) The performance of the "as-delivered" heat pump was measured. The machine's refrigerant circuit was instrumented with contact thermocouples and pressure taps to obtain a preliminary indication of the machine's thermodymamic operating characteristics.

2) These "as-delivered" results were used to study improvements that could be made to the manufacturer's unit.

3) Several hardware modifications were made to the machine. The effect of each change was examined separately and together with the other changes in a series of retests.

All performance tests of the air-source operation were done in accordance with ASHRAE Standard 37-1978, "Methods of Testing and Rating Unitary Heat Pump Equipment" using the Indoor Air Enthalpy Method (2). The same techniques were used to test the gas-source operation, except that the consumption of the unit was taken as the sum of the electric and fuel inputs. The indoor return air was maintained at a temperature of 21.1^{o}C and outdoor relative humidity was kept at 80% (3). Other test parameters (indoor-air flow rate, outdoor air temperature) were varied depending on the particular test. A more detailed description of the test apparatus and method is given in Appendix A. A schematic of the heat pump's major components and their interconnection is shown in Figure 1 along with an indication of the measurement points attached to the refrigerant circuit in the laboratory.



ACC	Accumulator
COMP	Compressor
CT	Capillary Tube
DI	Distributor
FD	Filter-Drier
IC	Indoor Coil
MU	Muffler
OC	Outdoor Coil
RV	Reversing Valve
GB	Gas Burner

A	Compressor Discharge
B	Condenser Inlet
C	Liquid Line
D	Evaporator Coil Traverse
E	Evaporator Outlet
F	Compressor Suction
□	Pressure Taps
○	Surface Thermocouples

Figure 1 Schematic of "as-delivered" Kool-Fire. Measurement locations added in the laboratory are given by the single letter designations. Refrigerant flow directions are shown for the heating mode; a "+" indicates flow away from the reader and a "." indicates the opposite.

2 "AS-DELIVERED" PERFORMANCE EVALUATIONS

2.1 PERFORMANCE TESTS

The steady state performance of the "as-delivered" heat pump was determined in both the gas and heat-pump modes at four different outdoor temperatures (8.3, 0, -8.3, -18 °C). Indoor air flow was maintained at about 515 L/s (1100 cfm) at a static pressure of 50 Pa (0.2 in. water gauge). These conditions were selected by the manufacturer as being appropriate for the unit he supplied and installed in our laboratory. However, a series of tests closer to ARI conditions (3) revealed little difference in performance at higher air flows and lower static pressures (shown later as part of Appendix B).

Bottled propane was used initially as the gas mode fuel, as was the case for the previous study (1). However, subsequent performance tests were done using natural gas, showing that there is no significant difference between the use of propane or natural gas as the fuel for the Kool-Fire unit.

The heat pump was instrumented with "line-piercing" pressure taps and contact thermocouples at selected points on the refrigerant piping (see Figure 1) to obtain a preliminary indication of the system's thermodynamic operating characteristics. Although insertion thermocouples give a more accurate reading of vapour-state conditions, contact thermocouples were used for the as-delivered tests so the system's integrity would not be affected by breaking into the circuit. Pressure was measured using a calibrated pressure transducer connected through a manifold system to the various pressure taps. Results revealed ways in which the heat pump performance could be improved. Potential improvements are discussed individually in the next section.

2.2 RESULTS AND DISCUSSION

Consumptions and Efficiencies

Table 1 gives the measured energy consumptions and efficiencies of the "as-delivered" unit operating in its heating modes. In pure heat pump mode, the unit uses no gas, and efficiency is expressed as a coefficient of performance (COP). Efficiency and COP values for the original 32HB model, as taken from reference (1), are shown in brackets for comparison purposes.

The gas-mode efficiency averaged 95%, about the same as for the original model. However, there has been a significant heat pump performance improvement (from a COP of 2.0 to 2.4 at 8.3°C). Figure 2 shows the variation of COP and capacity with outdoor temperature for the "as-delivered" machine in its heat pump mode. These curves show the normal decline in performance with outdoor temperature that is characteristic of all air-source heat pumps. Bear in mind that the HC3 does not normally operate in heat-pump mode at the lower outdoor temperatures; these curves are presented only to characterize the unit's thermodynamic behaviour. This test does not include defrost-cycle penalties because the unit is not intended to run at frosting temperatures. Note, however, that the unit is quite capable of acting as a heat pump at temperatures far below freezing.

Table 1 includes columns for both operating modes of the Kool-Fire heat pump at each test temperature. In addition, at the two standard test temperatures (+/-8.3°C), results for both propane-burning and natural-gas-burning condensing furnace modes are shown. For consistency and convenience, input and output quantities are given in kilowatts for all fuels and capacities. Fuel input values are based upon the higher heating value of the gas as tested. Within the limits of error on the efficiency measurements, condensing-gas-furnace-mode performance is independent of outdoor temperature and fuel type.

Table 1

STEADY STATE PERFORMANCE OF "AS-DELIVERED" KOOL-FIRE HEAT PUMP

Manufacturer's supplied refrigerant charge: 4.2 kg of R22.
Abbreviations shown under each outdoor temperature condition indicate Kool-Fire mode of operation:
"Prop" = Condensing gas furnace mode using bottled propane as the fuel.
"Gas" = Condensing gas furnace mode using natural gas as the fuel.
"Elect" = All-electric heat pump mode of operation.

Measured Value	Units	Outdoor Temperature and Conditions									
		8.3°C			0°C		-8.3°C			-18°C	
		Prop	Gas	Elect	Prop	Elect	Prop	Gas	Elect	Prop	Elect
Indoor Air Flow	litres/s	508	511	517	506	518	515	504	514	514	516
Temperature Rise Across Indoor Coil	°C	23.7	22.6	13.5	23.1	10.1	23.4	22.2	8.3	23.5	5.6
Electric Input	kW	3.68	3.61	3.47	3.69	3.29	3.72	3.75	3.31	3.70	3.16
Fuel Input	kW	11.3	10.5	-	11.9	-	12.1	10.8	-	11.6	-
Total Input	kW	15.0	14.1	3.47	15.6	3.29	15.8	14.5	3.31	15.3	3.16
Calculated Output	kW	14.4	13.7	8.50	15.0	6.30	14.6	13.8	5.20	14.5	3.50
Coefficient of Performance		.97	.97	2.4	.91	1.9	.92	.95	1.6	.95	1.1
(compared to prev. model tests (1))		(.98)		(2.0)			(.94)		(1.4)		(1.1)

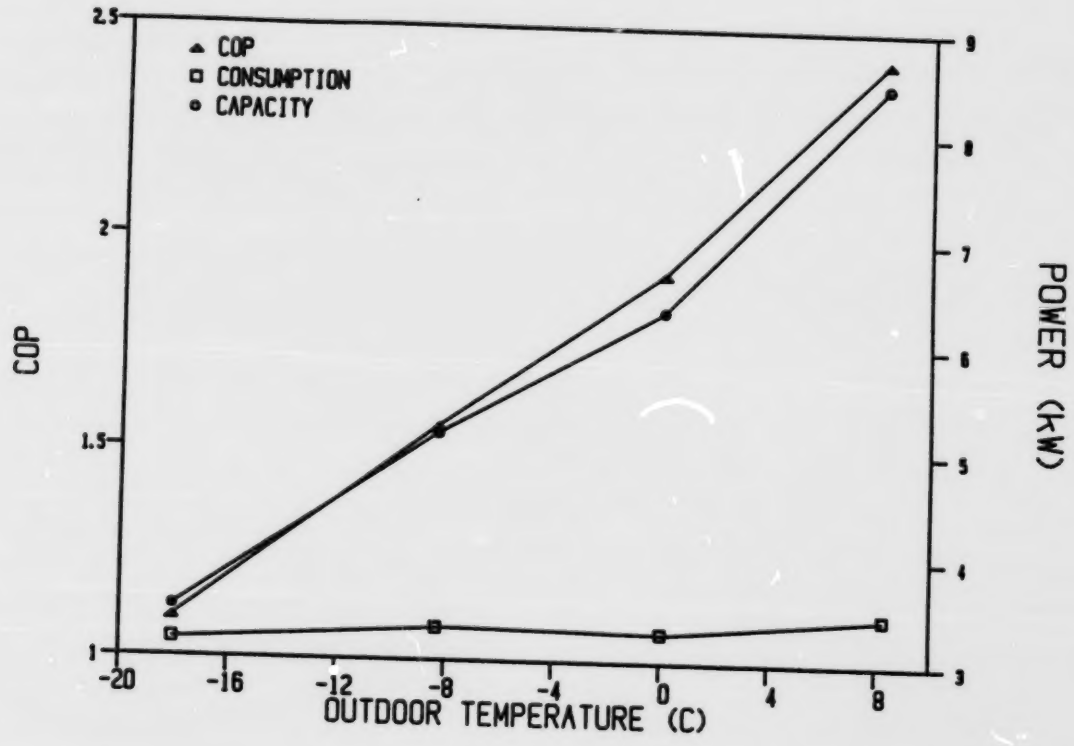


Figure 2 Temperature-Dependent Operating Characteristics of the "As-Delivered" Kool-Fire HC3 in Heat Pump Mode

Table 1 reveals another noteworthy characteristic of the Kool-Fire system. When the unit is operating as a condensing gas furnace, it still derives a significant portion (24%) of its input from the electric supply. You might think of the furnace mode as one that uses a "blend" of fuels that is mostly gas, but partly electricity. Thus, any assessment of operating cost of the unit must take into account the cost of this fuel blend.

Part of the electric consumption of the Kool-Fire heat pump, and indeed

any heat pump, is the power consumption of its fans and blowers. In the Kool-Fire unit, both the indoor and outdoor air movers are blowers. The individual consumptions of these blowers were measured as sub-components of the electric input in Table 1. The results are discussed in the next section.

Air Movers

The electrical consumption of the unit's outdoor air mover is uncharacteristically high, averaging 650 Watts. This is caused by the manufacturer's use of a belt-drive centrifugal blower rather than the more-commonly-used propeller fan in the outdoor unit. The blower consumption decreased under frosting conditions and increased slightly as the outdoor temperature went down. (Normal blower consumption was about 610 Watts at 8.3°C, while the highest consumption seen was about 720 Watts at -18°C.) This characteristic works against the unit's efficiency as a heat pump because the blower becomes an even-larger component of the system's total consumption at the very temperatures that reduce total output. A typical heat pump's outdoor propeller fan uses about half the power of this unit's blower. In general, propeller fans are more efficient than blowers when operating against the low static pressures found in most outdoor units (4).

Average indoor-blower consumption was fairly constant at about 270 Watts, at a discharge duct static pressure of 0.2 in. water gauge and an indoor air flow rate of 520 L/s (1100 cfm). Blower consumption in later tests averaged 470 Watts at a discharge head of 0.15 in. water gauge and an air flow rate of 700 L/s (1490 cfm). These consumptions are quite reasonable for a forced-air blower used in residential heating. (ARI Standard 240 (3) stipulates that calculations assume an indoor-air-mover consumption of 780 Watts for every 1000 L/s of air flow when testing a heat pump that doesn't include its own blower.)

Table 2

THERMODYNAMIC OPERATING CHARACTERISTICS OF "AS-DELIVERED" KOOL-FIRE

[Metric Units] Description (with key on Fig. 1)	Quantity	Outdoor Temperature and Conditions 8.3°C Electric	-8.3°C Electric	8.3°C N.Gas	-8.3°C N.Gas
Compressor Discharge (Point A)	P	1437	1259	1978	1979
	TS	38	32	51	51
	TA	50	52	65	64
	SH	12	20	14	13
Condenser Inlet (Point B)	P	1439	9871	1893	1900
	TS	37	23	49	49
	TA	43	40	60	60
	SH	6	17	11	11
Liquid Line (Point C)	P	644	461	991	999
	TS	8	-2	23	23
	TA	7	-4	22	22
	SH	-1	-2	-1	-1
Evaporator Traverse (Point D)	P	507	331	857	866
	TS	1	-12	18	18
	TA	1	-12	18	18
	SH	0	0	0	0
Evaporator Outlet (Point E)	P	457	306	748	751
	TS	-3	-14	13	13
	TA	-3	-14	13	13
	SH	0	0	0	0
Compressor Suction (Point F)	P	409	274	663	669
	TS	-6	-17	9	9
	TA	-5	-16	10	10
	SH	1	1	1	1

[Imperial Units]		47°F Electric	17°F Electric	47°F N.Gas	17°F N.Gas
Compressor Discharge (Point A)	P	211	183	287	287
	TS	100	90	123	123
	TA	121	125	148	147
	SH	21	35	25	24
Condenser Inlet (Point B)	P	209	143	275	276
	TS	99	73	120	120
	TA	110	104	140	139
	SH	11	31	20	19
Liquid Line (Point C)	P	93	67	144	145
	TS	47	28	74	74
	TA	45	26	72	72
	SH	-2	-2	-2	-2
Evaporator Traverse (Point D)	P	74	48	124	126
	TS	33	11	64	65
	TA	34	11	64	65
	SH	1	0	0	0
Evaporator Outlet (Point E)	P	66	44	109	109
	TS	27	7	56	56
	TA	27	7	55	56
	SH	0	0	-1	0
Compressor Suction (Point F)	P	59	40	96	97
	TS	21	1	48	49
	TA	23	3	49	50
	SH	2	2	1	1

P = Pressure (kPa, psia) TA = Actual Temperature (°C, °F)
TS = Saturation Temperature (°C, °F)
SH = Degrees of Superheat (°C, °F, -ve denotes Subcooling)

System Refrigerant Thermodynamics

Table 2 gives the "as-delivered" refrigerant temperatures and pressures at selected locations throughout the system in both heating modes at standard outdoor temperatures (+/-8.3°C). Natural gas was used as the gas mode fuel and the manufacturer's designated charge of R22 was in the system. To aid the reader in understanding the results, each reading is accompanied by a listing of the saturation temperature corresponding to the measured pressure, and a statement of the degree of superheating or subcooling present in the refrigerant at that location. (Table 2 is provided in both Metric and English versions for convenience.)

Low pressure drop through the evaporator, and from the evaporator outlet to the compressor inlet, is essential in the design of an efficient heat pump. If the return side pressure drop exceeds a few psi (a few tens of kPa) the compressor suffers from reduced capacity at a given outdoor temperature.

The "as-delivered" Kool-Fire heat pump suffers from high return-side pressure drop. This can be seen in the drop from point D to E for the 8.3°C heat pump mode test. This drop is 50 kPa (7 psi) through only part of the evaporator. The drop from point E to F is 48 kPa (7 psi) through the remaining return piping. The distributor at the inlet to the evaporator makes it difficult to evaluate the whole evaporator pressure drop directly, but point D is about 1/4 of the way through the coil, allowing the total drop to be estimated at 67 kPa (10 psi). A good choice of evaporator circuiting configuration would limit this drop to about 14 kPa (2 psi).

About 30% of the return piping's pressure drop is due to the filter-drier and the remainder to the piping and reversing valve. It is unusual to see a suction line filter in a heat pump because of the inevitable pressure drop it will cause.

Table 2 also shows a complete absence of superheat at the outlet of the evaporator in both high- and low-temperature, heat-pump-heating-mode tests. This is an unusual situation for a capillary-tube controlled machine because one normally expects some suction superheat to protect the compressor from flooding. In this system, the accumulator accomplishes this task, but it is still a sign of overcharging that no superheat is present.

The reader should recall that a capillary-tube system essentially controls the refrigerant conditions leaving the condenser. Any variation in refrigerant charge therefore appears on the low side of the system without affecting high-side conditions very much. Charge requirements vary with outdoor operating temperature and the machine needs the most charge at higher temperature conditions. Table 2 reveals no sign of charge insufficiency even at the higher operating conditions, leading us to suspect overcharging. Confirmation of this condition is presented in the next section.

The system was charged by the manufacturer at the time of installation. The recommended charging method simply involves adding refrigerant, while running in the gas mode, until the temperature of the top and bottom of the compressor shell were prescribed values (Temperatures: Oil Sump 43°C, Suction Gas 18°C). In this way the manufacturer feels that the compressor is being adequately cooled, especially in the high temperature gas and air-conditioning modes. This method resulted in a system charge of 4.2 kg (9.3 lbs) (see Appendix B).

3 STUDY OF POTENTIAL IMPROVEMENTS

3.1 DETERMINATION OF AN OPTIMAL CHARGE

Before modifications were made to the machine, the extent and significance of the potential refrigerant overcharge was investigated. As noted previously, the manufacturer's charge was determined to be 4.2 kg using the procedure outlined in Appendix B.

The machine was run in heat pump mode at 8.3°C and refrigerant was removed in 250 g steps. The COP and capacity increased to 2.71 and 9.35 kW respectively while the consumption decreased slightly to 3.45 kW at a charge of 2.7 kg (6 lbs). Table 3 and Figure 3 show these dramatic improvements in performance as the charge was gradually reduced. A detailed discussion of the test procedure and results is given in Appendix C.

The results of Appendix C, and some later work carried out at even lower charge levels, showed that a reduced refrigerant charge can increase the heating-mode performance without affecting gas-mode performance. However, the manufacturer worried that a reduced charge could adversely affect compressor-motor cooling and lubrication, as well as cooling-mode performance. In an extension to the original project, further testing (Appendix D) showed that a reduced charge did not overheat the motor. However, a somewhat higher charge of 2.95 kg (6.5 lbs) produced the best cooling performance as opposed to 2.25 kg (5.0 lbs) for the very best heating performance. Therefore, a charge of 2.5 to 2.7 kg was selected as being optimum for overall performance in both the heating and cooling modes.

Table 3
THE EFFECT OF REFRIGERANT CHARGE ON "AS-DELIVERED" PERFORMANCE
(Heat Pump Mode at 8.3°C)

Measured Quantity	Units	Total Charge (kg) 4.2	3.95	3.7	3.45	3.2	2.95	2.7	2.45
Indoor Air Flow	litres/s	532	532	531	531	529	529	529	529
Temperature Rise Across Indoor Coil	C	13.5	13.8	14.2	14.0	14.3	14.5	14.6	14.7
Electric Input	kW	3.52	3.54	3.55	3.54	3.54	3.49	3.45	3.42
Calculated Output	kW	8.67	8.88	9.09	9.00	9.14	9.30	9.35	9.38
Coefficient of Performance		2.47	2.51	2.56	2.54	2.58	2.66	2.71	2.74

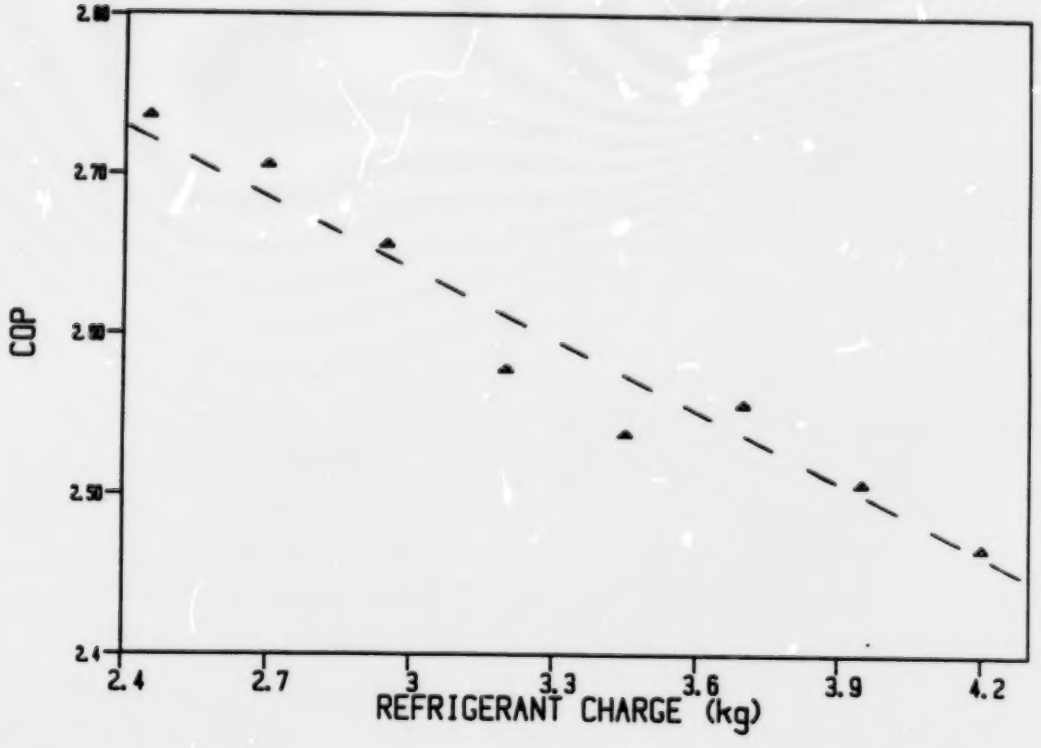


Figure 3 Performance of "as-delivered" Kool-Fire at various levels of refrigerant charge.

3.2 EXPANSION DEVICE IMPROVEMENTS

Since improvements in performance were found to be strongly dependent on refrigerant charge, a logical next step was to investigate any effect that a change in expansion device would bring. This is not an easy hardware modification task because of the complex arrangement of the Kool-Fire's capillary tubes (see Figure 1). The indoor coil has four separate capillary tubes, one on each circuit of the heat exchanger. An additional pressure drop is developed across another four tubes leading from the distributor on the outdoor coil.

A temporary piping arrangement as described in Appendix E was fitted to bypass the indoor section's capillary tubes and replace them with a hand expansion valve. In a series of steady-state, heat-pump-mode performance tests, no improvement in system COP could be obtained by either increasing or decreasing the refrigerant flow in the system, at the optimal charge levels previously determined. It is possible that another series of charge-optimization tests could have delivered higher heat pump performance with some other charge at these altered refrigerant flows, although it would be difficult to demonstrate this conclusively without extensive hardware modifications that would have included four new balanced capillary tubes or expansion devices.

There is some indication from the more extensive thermodynamic measurements (that will be reported later in Table 6) that the four capillary tubes are slightly oversized for proper condenser subcooling in the heat pump mode. A more restrictive set of capillaries would give more subcooling and deliver more output for the same input energy. However, it is recognized that the capillary tubes are about right for the gas mode conditions which are characterized by higher refrigerant flow rates. It is difficult to get one set of capillary tubes to perform well over so wide a range of heating- and cooling-mode conditions. Many manufacturers use two different sets, one for heating and one for cooling. Another type of control valve might do a better job in the

complexity would be increased. Suffice it to say that the present set of capillary tubes is probably a compromise, but no great loss of performance can be attributed to this one factor. Other control devices such as thermostatic expansion valves, subcooling control valves, or electrically-controlled expansion valves would, however, yield a machine that is much more tolerant of refrigerant charge errors and variations (4).

3.3 REFRIGERANT COMPONENT SELECTION AND CONFIGURATION

The components selected for the Kool-Fire appear to be of high quality and their assembly has been executed in a simple manner. For instance, a premium-quality compressor is employed in the unit and no quick-connect couplings are present, thus avoiding unnecessary pressure drops in the system. However, as shown in Table 2 and discussed in the last section under system refrigerant thermodynamics, the particular piping circuit selected for the outdoor coil results in a significant refrigerant-side pressure drop during heat-pump heating mode. This was verified at 74 kPa (10.7 psi) in later testing (Table 5), whereas good evaporator design would limit the pressure drop to about 14 kPa (2 psi).

Such a pressure drop is of little consequence in the fuel-burning mode of operation, because heat flow is essentially "down hill" with no reliance on the reverse Rankine thermodynamic cycle for the delivered heat output. However, in heat pump mode, each extra pound of pressure drop on the suction side of the system lowers the output capacity and COP by about 1%. For this reason, much attention is paid to any pressure drop through the reversing valve and suction piping of a high-efficiency heat pump. Pressure drop through the evaporator may not be quite as significant for a heat pump's performance, but it is estimated that about a 5% improvement is possible by re-designing the Kool-Fire's evaporator. The present evaporator has four parallel paths through it and the pressure drop could be reduced by simply changing the circuitry to have more parallel paths.

Such a major change falls outside the scope of this project, and is left to future investigators for implementation.

Another source of suction pressure drop that has already been mentioned is the suction-line filter/drier used on the Kool-Fire heat pump. If this component can be replaced by the more common liquid-line filter/drier, then another pressure drop (of about 2 psi) can be avoided. In the liquid-line position, a smaller bi-directional filter would be used and its pressure drop would be of no conseqence because of the normal pressure change along the liquid line and through the expansion device.

As part of the hardware modifications performed on the Kool-Fire heat pump, the filter/drier was removed from the suction line and replaced with a liquid-line filter. Results of this re-test are reported in Section 4.

3.4 SELECTION OF AN IMPROVED OUTDOOR AIR MOVER

As previously mentioned, most contemporary heat pumps use a propeller fan as the outdoor air mover because the back pressure or head loss across the outdoor coil is usually quite small. However, on this machine the outdoor cabinet has sheet metal flaps on both the air inlet and outlet. This is part of the manufacturer's method of protecting the gas flame from wind gusts and heat loss. This unusual configuration produces a relatively high air-side pressure drop and forces the manufacturer to use a centrifugal blower to open these flaps during the heat pump mode.

Calculations in Table 4 show the performance improvement that could be realized if the centrifugal blower could be replaced with a propeller fan. A typical outdoor-fan consumption of about 300 Watts was assumed, based upon tests of a similarly-sized heat pump (4). The values used for total output and consumption with a blower in Table 4 correspond to

measurements made of the improved Kool-Fire heat pump as tested later in Section 4. The improvements shown with the substitution of a fan are hypothetical, and are carried through to indicate the potential effect of such a design change.

The bottom line shows high- and low-temperature-heat-pump-mode COP's of 3.1 and 2.1 respectively. These would be commendable for any heat pump, even by contemporary standards, and they serve to show that there are no real flaws in the design of the Kool-Fire's refrigerant system. Of course, to accommodate the fan, the cabinet would have to be re-designed to eliminate the high static pressure loss while still protecting the flame from the weather when operating in the gas mode.

Table 4

HYPOTHETICAL IMPROVEMENTS ACHIEVED BY REPLACING THE OUTDOOR BLOWER WITH A FAN

[Heat Pump Mode]	Units	Outdoor Temperature		
		8.3°C	0°C	-8.3°C
Blower Consumption	Watts	650	620	680
Fan Consumption*	Watts	300	300	300
Savings with Fan*	Watts	350	320	380
Total Output	Watts	9880	7580	5850
Consumption with Blower	Watts	3520	3390	3120
Consumption with Fan*	Watts	3170	3070	2740
COP with Blower		2.8	2.2	1.9
COP with Fan*		3.1	2.5	2.1

4 HARDWARE MODIFICATION AND IMPROVEMENT VERIFICATION TESTING

4.1 DISCUSSION

The initial testing reported in Section 2 was carried out without altering the manufacturer's heat-pump design in any way. After establishing the machine's basic characteristics, it was felt that more information could be gained by physically altering some of the refrigerant-circuit layout. Strictly speaking, the resulting heat pump cannot be equated with the original device because of the change in configuration. However, care was taken to ensure that the basic properties of the original heat pump remained unchanged and were, in fact, repeatable under conditions that matched the "as-delivered" testing previously reported.

The altered heat pump provided additional information in three areas:

- thermodynamic measurement techniques were improved by 1) replacing contact-type thermocouples with ones that were inserted directly into the gas streams through special thermo-wells added to the system, and 2) constructing pressure access points at previously inaccessible locations such as the outlet of the indoor coil ahead of the capillary tubes and the inlet of the outdoor coil after the distributor tubes.

- one of the suction pressure drops identified previously was eliminated by removing the suction-line filter/drier and replacing it with a filter/drier in the liquid line.

- the capillary tubes were isolated and paralleled with a hand-operated expansion valve for the purpose of investigating different refrigerant flows.

A detailed description of the work carried out is given in Appendix E.

The modified Kool-Fire heat pump was also used for the bulk of the charge-optimization tests in heating and cooling mode and for a series of motor-winding and discharge-temperature tests carried out to verify that the reduced charge does not seriously stress the compressor (Appendix D).

4.2 PERFORMANCE RESULTS

Performance results in electric heat pump mode for the modified Kool-Fire are shown in Table 5 and Figures 4 and 5. As mentioned in Section 3.1 and detailed in Appendix D, the very best heat-pump performance at 8.3°C is achieved with a charge of 2.25 kg of R22. However, for good air-conditioning performance and less-stressful operating temperatures, a

Table 5
HEATING MODE PERFORMANCE OF IMPROVED KOOL-FIRE

		Outdoor Temperature and Conditions				
Measured Quantity	Units	Elect 8.3°C	Elect 8.3°C	Elect 0°C	Elect -8.3°C	Gas -8.3°C
Refrigerant Charge	kg	2.7	2.25	2.25	2.25	2.9
Indoor Air Flow	litres/s	718	717	713	722	704
Temperature Rise Across Indoor Coil	°C	11.3	11.4	8.8	6.7	16.2
Electric Input	kW	3.66	3.52	3.39	3.12	3.52
Natural Gas Input	kW					11.13
Total Input	kW	3.66	3.52	3.39	3.12	14.65
Calculated Output	kW	9.79	9.88	7.58	5.85	13.84
Coefficient of Performance		2.7	2.8	2.2	1.9	.94

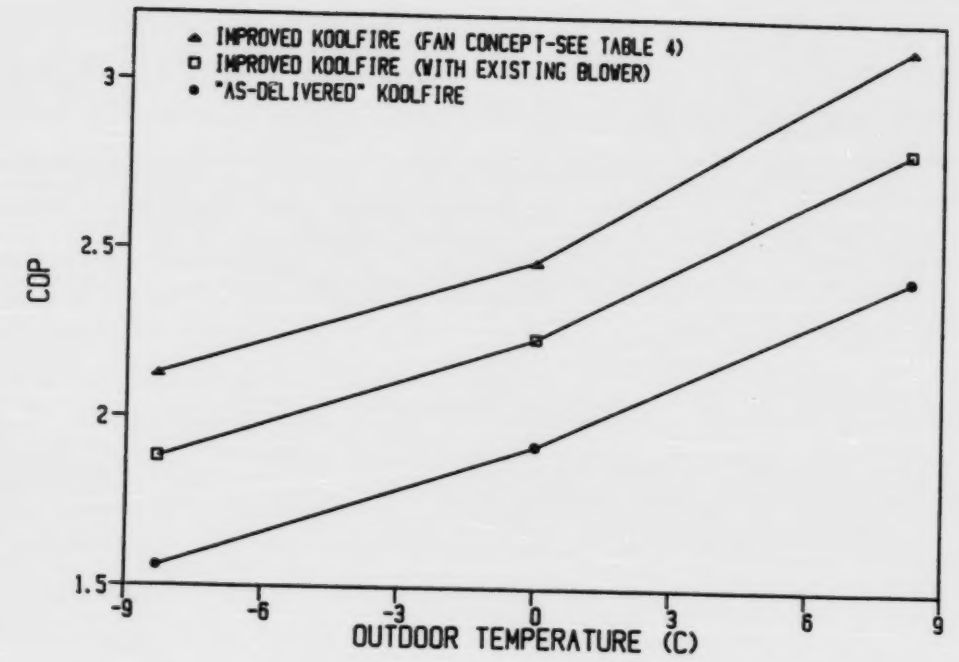


Figure 4 Comparison of heat-pump-mode performance for the "as-delivered" Kool-Fire and the improved Kool-Fire with and without an outdoor fan.

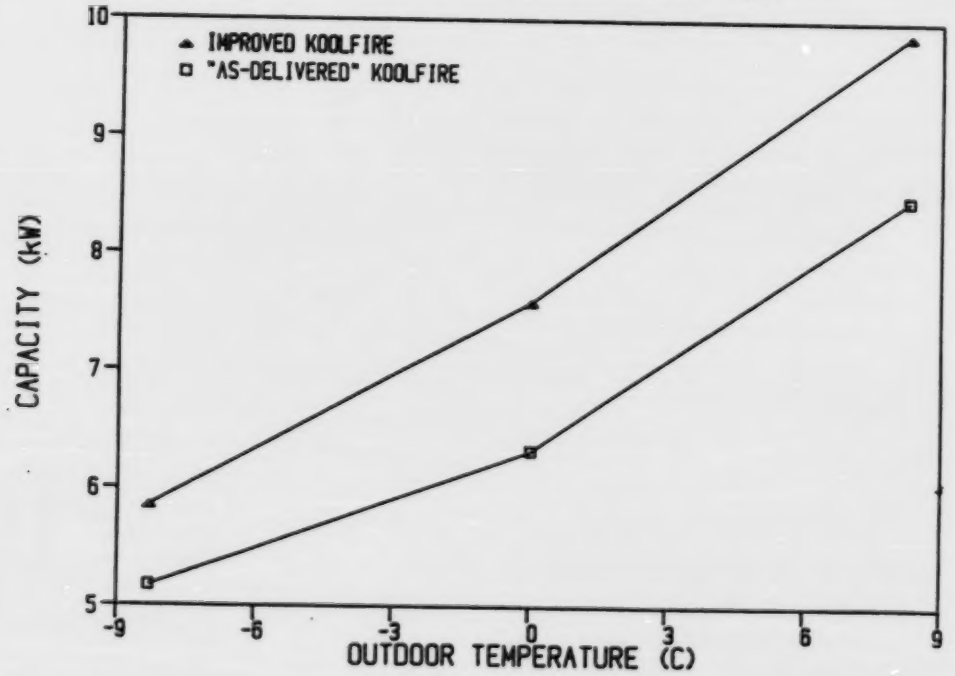


Figure 5 Comparison of heat-pump-mode capacity for the "as-delivered" Kool-Fire and the improved Kool-Fire

charge of 2.7 to 2.9 kg is recommended. Results in Table 5 cover both levels of charge at 8.3°C and the difference in heating COP is slight (0.1). A single gas-mode test is included to show that this mode's performance remains characteristically high at the reduced charge.

Figures 4 and 5 present the heat-pump-mode performance results for the improved Kool-Fire. For comparison purposes, the "as-delivered" performance curves are included to show the difference that charge optimization makes to the system. Remember that these Figures show results at a charge of 2.25 kg (5.0 lbs), the condition that produced maximum COP and output in both high- and low-temperature-heating-mode tests. These tests reflect any improvement that removing the suction-line filter/drier has had, but do not include any potential improvement that could be obtained by re-designing the outdoor coil for lower refrigerant-side pressure drop.

Inspection of Figure 4 shows that the modified Kool-Fire achieves steady-state-heat-pump-mode COP's that are quite acceptable by today's standards. With the exception of a few "trophy models", heat pumps rarely exceed COP values of 3.1 and 2.1 at the high- and low-temperature test conditions, respectively (+/-8.3°C or 47 & 17°F). The "as-delivered" Kool-Fire is about 30% low by this criteria, but the modified machine has only about a 10% gap to go before equaling current machines. However, most if not all of this gap is eliminated if the hypothetical outdoor blower improvement discussed in Section 3.4 is factored in as shown in the upper line on Figure 4.

4.3 THERMODYNAMIC MEASUREMENTS

The thermodynamic operating characteristics of the modified Kool-Fire heat pump are shown in Table 6. (Table 6 is divided into parts "a" and "b" for metric and imperial units respectively.) Comparisons can be made with the results of the "as-delivered" machine shown in Table 2.

Table 6a

THERMODYNAMIC OPERATING CHARACTERISTICS OF MODIFIED KOOL-FIRE

[Metric Units] Description (with key on Fig. E1)	Quantity	Outdoor Temperature and Conditions				
		8.3°C Electric	8.3°C Electric	0°C Electric	-8.3°C Electric	-8.3°C N.Gas
Charge	kg	2.7	2.25	2.25	2.25	2.9
Compressor Discharge (Point A)	P	1465	1398	1311	1199	1774
	TS	38	36	34	30	46
	TA	64	94	67	72	79
	SH	26	58	33	42	33
Condenser Inlet (Point B)	P	1386	1331	1267	1172	1657
	TS	36	34	32	29	43
	TA	56	86	56	58	76
	SH	20	51	24	29	32
Condenser Outlet 1 (Point C)	P			1275		1644
	TS			32		43
	TA			29		33
	SH			-3		-10
Condenser Outlet 2 (Point C)	P	1372	1321	1252	1164	1640
	TS	36	34	32	29	43
	TA	30	30	29	28	34
	SH	-6	-4	-3	-1	-9
Evaporator Inlet 1 (Point D)	P			449		811
	TS			-3		16
	TA			-3		16
	SH			0		0
Evaporator Inlet 2 (Point D)	P	559	537	441	357	804
	TS	4	2	-4	-10	15
	TA	4	2	-4	-10	15
	SH	0	0	0	0	0
Evaporator Outlet (Point E)	P	476	463	378	307	679
	TS	-1	-2	-8	-14	10
	TA	1	6	-6	-13	13
	SH	2	8	2	1	3
Compressor Suction (Point F)	P	429	426	344	283	611
	TS	-5	-5	-11	-16	6
	TA	-4	2	-10	-16	7
	SH	1	7	1	0	1

P = Absolute Pressure (kPa) TS = Saturation Temperature (°C)
TA = Actual Temperature (°C) SH = Degrees of Superheat (°C)
(-ve denotes Subcooling)

Table 6b

THERMODYNAMIC OPERATING CHARACTERISTICS OF MODIFIED KOOL-FIRE

[Imperial Units] Description (with key on Fig. E1)		Outdoor Temperature and Conditions				
		47°F Electric	47°F Electric	32°F Electric	17°F Electric	17°F Gas
Charge (lb)		6	5	5	5	6.5
Compressor Discharge (Point A)	P	212	203	190	174	257
	TS	101	97	93	86	115
	TA	148	201	152	161	175
	SH	47	104	59	75	60
Condenser Inlet (Point B)	P	201	193	184	170	240
	TS	97	94	90	85	110
	TA	133	186	133	136	168
	SH	36	92	43	51	58
Condenser Outlet 1 (Point C)	P			182		238
	TS			90		109
	TA			84		92
	SH			-6		-17
Condenser Outlet 2 (Point C)	P	199	192	182	169	238
	TS	96	93	89	84	109
	TA	87	87	84	82	94
	SH	-9	-6	-5	-2	-15
Evaporator Inlet 1 (Point D)	P			65		118
	TS			26		60
	TA			26		60
	SH			0		0
Evaporator Inlet 2 (Point D)	P	81	78	64	52	117
	TS	38	36	25	14	59
	TA	38	36	25	14	59
	SH	0	0	0	0	0
Evaporator Outlet (Point E)	P	69	67	55	45	98
	TS	30	28	17	7	50
	TA	33	42	22	8	56
	SH	3	14	5	1	6
Compressor Suction (Point F)	P	62	62	50	41	89
	TS	24	24	12	3	43
	TA	25	35	13	3	45
	SH	1	11	1	0	2

P = Absolute Pressure (psia)
TA = Actual Temperature (°F)
TS = Saturation Temperature (°F)
SH = Degrees of Superheat (°F) (-ve denotes Subcooling)

Improvements can be seen in many key areas of performance in the heat pump mode while performance in the gas mode remains high. It should also be noted that the superheated-gas temperatures at the compressor suction and discharge are now much more accurate than the values in Table 2. (The surface thermocouples used to measure gas temperatures in the "as-delivered" heat pump are influenced by surrounding ambient temperatures, even when heavily insulated.)

At the lower charge involved in the "modified" testing, discharge pressures are equal to or less than those of the "as-delivered" results. Condenser pressure drops are shown by the new tests to be low, in sharp contrast to the evaporator pressure drop which is 83 kPa (12 psi) at the higher-temperature, heat-pump test condition.

Compressor suction pressures are consistently higher in the modified-heat-pump modes, and this results in higher compressor capacities. Most of this pressure increase is due to the removal of the former suction-line filter/drier. Evaporator superheat, which was missing in the "as-delivered" tests, is now shown to be strongly dependent on charge level (as expected in a capillary-tube system). These levels of superheat are quite acceptable, even in the gas mode, and are well worth the increased level of performance in the modified unit.

As mentioned earlier, there is some indication that condenser outlet conditions are not subcooled quite enough to achieve peak performance in the modified unit. Slightly more restrictive capillary tubes would give more subcooling, but of course a new optimum charge would then have to be determined.

4.4 RELIABILITY CONSIDERATIONS

The results presented in Section 3.1 and verified in this Section show that a reduced refrigerant charge can increase the heating mode

performance of the Kool-Fire heat pump without affecting gas mode performance. However, it is possible that a reduced charge could adversely affect either compressor-motor cooling or air-conditioning-mode performance. Inadequate compressor cooling would be reflected in high discharge temperatures, and the values shown for all modes in Table 6 are well within manufacturer's guidelines. In general, manufacturers are not concerned if discharge temperatures on R22 hermetic compressors do not exceed 100°C (212°F) on a continuous basis. In fact, one manufacturer specifies acceptable temperatures as any value under 105°C, measured 150 mm from the compressor shell on the surface of the discharge line. Another manufacturer specifies that a high-temperature, cut-off control in this location be set to 235°F (113°C). The temperatures measured in Table 6 are therefore very conservative because they are measured with direct-insertion thermocouples.

The influence of charge on cooling-mode temperatures and performance was also investigated in light of any possible concern over the suggested charge reduction. The results presented in Appendix D demonstrate that although discharge temperatures are higher in cooling mode, they are not above manufacturer's recommendations even at the highest outdoor-test temperatures, and lowest charge conditions. As Appendix D shows, even the cooling-mode efficiency of the Kool-Fire heat pump benefits from a reduced charge.

As a final and direct check on the internal temperature levels experienced by the compressor motor during reduced-charge operation, a modified, Sealey, winding-temperature apparatus was employed to sense the condition of the motor during actual operation of the Kool-Fire unit. The results (see Appendix D) show that at no time during operation at any of the reduced-charge levels selected for optimum performance did the windings of the motor exceed 55°C. Although this is slightly warmer than was the case with the "as-delivered" unit's charge, it is well within acceptable conditions for winding insulation materials. In fact, the sharp drop in winding temperatures noticed in Figure D2 at the higher

charge levels may be an indication that liquid refrigerant is entering the suction side of the compressor with possible dilution of the lubricating oil as a consequent risk!

The conclusion of the investigation into the modified Kool-Fire is that a charge of about 2.5 to 2.7 kg would be optimum for overall performance in both the heating and cooling modes if the present capillary tube network is used. If some other refrigerant control device is used to provide either more versatile control in the various modes of operation or more tolerant behaviour under different charge conditions, then some other level of system charge may be appropriate. However, the performance achieved under this different control strategy will equal the optimum values deduced herein with the modified machine. Any further increase in performance will only come with a change in evaporator design, or the possible elimination of the unit's reversing valve (see next Section).

5 OPERATIONAL CONSIDERATIONS FOR "KOOL-FIRE"-TYPE HEAT PUMPS

The Kool-Fire is a new concept in residential heating and its performance in actual applications depends not only on the hardware's design, but also upon such application and operational considerations as correct sizing, appropriate air- to gas-source changeover, and possible defrost methods. Thus, there are several areas to consider, aside from hardware improvements, for the proper optimization of Kool-Fire-type heat pump systems.

5.1 OPERATING COSTS

COST OF FUEL BLEND

The Kool-Fire may be said to use a blend of fuels (one quarter electric and three quarters gas) which must be taken into account when determining the operating costs of the machine. Table 7 shows the cost of heat from the Kool-Fire in gas mode (2.86 c/kWh) compared to that of a conventional gas furnace (2.81 c/kWh) based on steady-state operating efficiencies at one particular combination of gas and electric rates. A similar calculation can be done for other rate combinations. Of course, actual heating costs will differ from these values due to cycling losses for both machines and, more significantly, to the free energy obtained during the Kool-Fire's air-source operation. Actual costs and other operating particulars would best be determined by field tests.

FIELD PERFORMANCE vs LAB PERFORMANCE

Normal laboratory tests are best suited for determining steady-state efficiencies and the ideal operation of a device. However, field tests are the best method of determining actual seasonal, operating costs and the long-term reliability of a device. Field tests are also the best way of determining site-related factors such as ease of installation,

Table 7

ENERGY MIX COMPARISON BETWEEN BIVALENT HEAT PUMP AND CONVENTIONAL GAS FURNACE

		Input Energy (kW)	Input Energy Fraction	Input Energy Cost (c/kWh)	Steady State Efficiency	Output Heating Energy (kW)	Output Energy Fraction	Final Cost of Heating (c/kWh)
Kool-Fire in Gas Mode	Electricity	3.525	.24	4.75	100	3.53	.25	4.75
	Gas	11.131	.76	2.08	93*	10.39	.75	2.24
	Total or Average	14.656	1.00	3.77	95	13.92	1.00	2.86
Conventional Gas Furnace	Electricity	.36	.01	4.75	100	.36	.02	4.75
	Gas	27.3	.99	2.08	75	20.48	.98	2.77
	Total or Average	27.66	1.00	2.11	75	20.84	1.00	2.81

* Computed value consistent with measured overall efficiency of 95% and obtained by taking the electric fraction at 100% efficiency.

quality of heat, and noise problems.

No attempt has been made here to analytically estimate any seasonal performance values based on the steady-state results reported herein. The great difficulty in doing this for the Kool-Fire, bivalent heat pump arises because no accurate split between its two operating modes can be deduced without further observations. This field work can be carried out effectively using an alternate-day or alternate-time-period method as described in reference 7. By comparing a known (all-electric) and an unknown heating system on alternate days in the same house over a long enough period of time, the seasonal operating efficiencies and costs of the unknown device can be accurately deduced.

5.2 CHANGEOVER METHOD

Another significant influence on the Kool-Fire's cost of operation is the selection of an appropriate control scheme to switch from air-source to gas-source operation. The Kool-Fire HC3 currently uses a thermostat located in the outdoor unit for this purpose. The thermostat has a setpoint of about 9°C with a 3°C differential setting. However, this switchover method is somewhat erratic. Other investigators (Electrical Testing Laboratories) (6) found that the actual changeover temperature could vary as much as 7°C either side of the thermostat setpoint.

This erratic behaviour may be understood by examining the mechanism involved in a switchover from air-source to gas-source. The thermostat has its sensing bulb located in a small passageway between the outdoor unit's blower compartment and the exhaust from the coil. Operating in the air source, the blower draws coil exhaust air across the sensing bulb. However, the exhaust air, having passed over the outdoor coil, may be 3 to 4 degrees below the ambient temperature.

As the coil exhaust air drops below the thermostat setpoint, the Kool-Fire switches to the gas mode. With the gas burner on and the

outdoor blower disabled, cool exhaust air is no longer being drawn through the passageway and the sensing bulb warms back up to the ambient temperature. The Kool-Fire switches back to the air-source operation and again the sensing bulb is cooled to the point where the gas mode comes back on. It is possible that at a particular outdoor temperature this process may be repeated several times during a heating cycle.

LOWER CHANGEOVER TEMPERATURE

Another concern with the current changeover method is the apparent loss of overall efficiency due to the relatively high switchover setpoint. By setting the changeover thermostat at 9°C, the manufacturer eliminated the need for a defrost cycle and started using the gas source at a relatively high temperature. The manufacturer also reasoned that since the gas-source efficiency was 95% and the output was 1.5 times the air-source output, he should take advantage of the gas-source operation as much as possible. However, as may be seen from the Table 7 cost comparison, and the tabulated performance results, higher energy efficiency and lower operating costs could be obtained by operating the air-source mode well below 0°C.

A quick calculation based on steady-state performance (Appendix F) shows the advantage of operating the air-source at lower outdoor temperatures. This calculation shows that the full heating requirements of the home can be met by the Kool-Fire air-source down to a balance point of -8°C. Below this temperature, the gas-source would be needed to supplement the air-source heat pump mode.

This control strategy would save the homeowner energy and money since the air-source delivers almost twice as much energy as it consumes even at this low temperature. It is apparent, however, that some method of defrosting the Kool-Fire's outdoor coil would need to be devised before the air-source could be used at lower outdoor temperatures.

NEW CONTROL METHOD

It may be possible to use the gas flame to defrost the outdoor coil, providing the frost on the coil does not impair the combustion of the gas. An appropriate sensor would sense frost on the coil and initiate the gas-source. Hot flue gases would melt all the frost and heat would still be delivered to the home. In fact, at this writing, the manufacturer has developed a new Kool-Fire model with an appropriate "gas-defrost" control strategy. The new model is designated "LTH".

The Kool-Fire LTH uses a two-stage indoor thermostat and a sensor on the inlet flap of the outdoor cabinet as the defrost initiator. The two-stage thermostat gives priority to the air-source heating, switching to the gas source when the house heating load cannot be met by the air-source capacity or when a defrost demand occurs. Once the gas source has been initiated either by the thermostat's second stage or by the defrost sensor, it stays on until the thermostat's first stage has been satisfied.

This new control scheme has several distinct advantages over the old scheme:

- Since the second-stage of the thermostat has effectively replaced the switchover thermostat, there are no more erratic switchovers.
- The Kool-Fire can now take advantage of its higher, air-source efficiency at lower outdoor temperatures since the gas source is not initiated until the house load exceeds the air-source capacity and even then, the Kool-Fire returns to air-source heat pump mode at the start of each new cycle.
- The Kool-Fire's unique gas-fired defrost places little stress on the compressor and continues to heat the home during a defrost, unlike the reverse-cycle defrost used with most residential heat pumps. Neither is there a need for a defrost terminator since the Kool-Fire simply switches to the higher-capacity gas source and continues running in this mode until the thermostat's first stage is satisfied.

5.3 BIVALENT OPERATION FROM A STORED FUEL

A home's yearly fossil fuel consumption with an improved Kool-Fire unit might conceivably be so low as to make connection to natural-gas lines uneconomical. Under such circumstances, installation in both rural and urban areas would be served best by a source of bottled propane. This becomes especially practical if the annual propane consumption is reduced to the point where only one small tank or a few manageable refills are needed per winter.

A hypothetical, yearly propane consumption may be calculated for the improved Kool-Fire operating as described in the previous section (-8°C balance point, two-stage thermostat, and gas-fired defrost). As described in Appendix F, the total amount of propane required over the heating season would be about 162 kg (358 lb) for a typical house in the Toronto area. Rural customers who use propane for water and space heating may already have large (1000 or 2000 lb) propane tanks. Thus, such a rural customer could instal a Kool-Fire without the additional cost of a tank and cut the amount of propane he uses for space heating by at least half.

5.4 MANUAL SUMMER/WINTER SWITCHOVER

Since the bivalent heat pump described in this report requires no shift of its reversing valve, except on a seasonal basis, it is possible that the common, electrically-activated reversing valve could be replaced with a manual valve system. Although this likely would not reduce hardware costs, there is some evidence (4) that automatic reversing valves can reduce heat-pump efficiency by about 5%. The energy savings produced by having a positive-isolating hand valve may be sufficient compensation for the owner to perform the spring and fall changeover manually.

6 CONCLUSIONS AND RECOMMENDATIONS

The Kool-Fire HC3 has improved significantly from a previously-tested model. In addition, there are several other areas which can be improved to bring the Kool-Fire up to its full performance potential. The highlights of our findings are listed below.

6.1 CONCLUSIONS

1. There has been a 20% improvement in air-source heat pump performance. For example, at an outdoor temperature of 8.3°C, a COP of 2.0 for a previously-tested Kool-Fire 32HB has risen to 2.4 for the current HC3 model. Gas-source efficiency remained unchanged at 95% and was insensitive to the fuel used - natural gas or propane.

2. An analysis of the Kool-Fire's thermodynamic characteristics showed that air-source performance was hindered by excessive evaporator- (70 kPa) and return-piping (50 kPa) pressure drops. To demonstrate the effect of the return-piping pressure drop, the filter in the suction line on the as-delivered machine was replaced by a filter in the liquid line. This change reduced the return-piping pressure drop by 14 kPa (2psi) resulting in an estimated 2% performance improvement.

3. A significant improvement to the machine's performance is possible by changing the refrigerant-charge/expansion-device combination. The manufacturer uses only a single set of capillary tubes as the expansion device for both the heating and cooling mode and the as-delivered charge was too great.

Preliminary results showed that performance increased significantly as refrigerant charge was reduced. The improvement was apparent as an increase in capacity and a reduction in compressor consumption. Peak performances for the machine with a 3.5 m (12 ft) line set were obtained with refrigerant charges of 2.25 kg (4.75 lbs) and 2.95 kg (6.5 lbs) for

the heating and cooling modes respectively. At this charge, no improvement in performance could be made by adjusting the expansion device, but it is possible that other charge and expansion-device combinations would perform even better.

The final, peak, heating COP's were 2.8 and 1.9 at 8.3°C and -8.3°C respectively. These values were achieved with a refrigerant charge of 2.25 kg while using the capillary tubes as the expansion device and with a liquid-line filter as opposed to the suction-line filter. A peak, cooling COP of 1.86 (6.3 EER) was obtained for the same system at standard rating conditions (26.7°C indoor and 35°C outdoor temperatures) and with a charge of 2.95 kg. No investigation of EER improvement with altered expansion devices was carried out. For the gas mode, charge reductions did not affect the efficiency but did reduce the compressor consumption slightly.

4. The electricity consumption of the outdoor air mover was found to be high for this size of residential heat pump, averaging 650 Watts. Unfortunately, the manufacturer has been forced to use a high-consumption blower due to the high, air-side pressure drop through the outdoor cabinet. This pressure drop is a result of having to design the cabinet with flaps and baffles to protect the gas flame from wind and weather.

A calculation was done to show the improvement that could be realized if the squirrel-cage blower could be replaced with a propeller fan. By using the propeller fan, the total electric consumption could be reduced by 350 Watts for this size of machine. This improvement would boost the peak COP we obtained from 2.8 to a respectable 3.1. Although this mechanical alteration may not be possible with the Kool-Fire's present configuration, the calculation serves to place the assessment on an equal footing with other heat pumps.

5. The Kool-Fire's overall seasonal performance could be greatly increased by using air-source heating down to lower, outdoor ambient

temperatures. Model HC3, tested here, uses a single-stage thermostat to activate the Kool-Fire while an outdoor thermostat switches from the air source to the gas source at about 9°C. Although the Kool-Fire eliminates the need for a defrost cycle by setting the switchover at this high temperature, the higher-capacity gas mode is fully employed at temperatures where the capacity of the air-source is still adequate to heat the home. The Kool-Fire passes up a lot of low-cost, air-source heating by using this control scheme.

6.2 RECOMMENDATIONS

1. The manufacturer should develop a specific, installation charging method similar to the one used by most heat-pump manufacturers. Instead of using a method which depends on rough operating parameters, the charge should be dependent on the size of machine and length of refrigerant piping used in the particular installation. By adopting this method the possibility of poor performance due to the incorrect refrigerant charge would be eliminated.

2. Performance could be improved further by reducing all line and coil pressure drops and by using separate expansion devices for heating and cooling. The return pressure drop could be reduced 14 kPa (2 psi) by removing the system filter from the suction line and placing it in the liquid line where the pressure drop is of no consequence because of the expansion-device pressure drop. Remaining line pressure drops could be reduced by increasing the number of coil circuits and by increasing the refrigerant piping diameter. The manufacturer has sacrificed some efficiency for simplicity and reliability by using a single set of capillary tubes as the heating and cooling expansion device. However, increased performance from using separate expansion devices would offset their higher initial cost. The best combination of expansion devices would probably be capillary tubes for cooling and some type of expansion valve for heating.

3. The air-source electric consumption might be reduced by replacing the outdoor blower with a propeller fan. A propeller fan could deliver the same air flow across the outdoor coil at half the consumption of the existing blower. This would boost the air-source COP by about 10%. However, to accommodate the propeller fan the outdoor cabinet would have to be re-designed to reduce the air-side pressure drop while still protecting the gas flame from the wind and weather.

4. The overall seasonal efficiency could be improved by adopting a control strategy that would use more air-source heating and less gas-source heating. Indeed, a new control scheme, that has already been adopted by the manufacturer, uses a two-stage thermostat which activates the air and gas source on the first and second stages respectively. Thus, the gas source replaces the air source only below the temperature at which the house load exceeds the air-source capacity. The outdoor coil, which now must run during frosting conditions, may be defrosted by activating the gas flame and allowing the hot flue gases to melt the frost.

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APPENDIX A

TEST METHOD AND EQUIPMENT

PERFORMANCE TESTS

All performance tests were done in accordance with ARI Standard 240-81: "Air-Source Unitary Heat Pump Equipment" (3). The procedure and apparatus are described in ASHRAE Standard 37-'978: "Methods for Rating Unitary Air-Conditioning and Heat Pump Equipment" - Indoor Air Enthalpy Method (2). This method uses a calibrated measurement loop to measure the air flow rate and air temperature difference across the indoor coil of the test equipment. The heat transferred by the heat pump to or from the indoor air can then be determined.

STANDARD RATING CONDITIONS (2)

Heating - 80% Relative Humidity Outdoors
21.1°C Indoor Temperature
8.3 and -8.3 °C Outdoor Temperatures

Cooling - 50% Relative Humidity Indoors
27.6°C Indoor Temperature
35°C Outdoor Temperature

In addition to these standard conditions, the Kool-Fire was tested at outdoor temperatures of -18°C and 0°C in the heating mode. The tests at 0°C give a good indication of the machine's performance in frosting conditions while the test at -18°C shows how the performance suffers at very cold temperatures.

HEATING TESTS

Measured Quantities: The following quantities were measured every ten

minutes over a one hour interval with the heat pump operating at steady state. (Symbols used are summarized at the end of this Appendix.)

1) E_e - Total electric energy consumed - the sum of the energy consumed by the compressor, indoor fan, outdoor fan, and all heat pump controls,

2) E_g - Total gas energy consumed when operating in the gas mode,

3) Δp - Pressure drop across the two air flow measurement nozzles,

4) mV - The voltage produced by a 40 junction thermopile indicating the temperature difference across the indoor coil,

5) W_i - return air humidity ratio,

6) T_d - discharge temperature at the thermopile,

Equations: The total energy transferred to the air (q_{thi}) by the coil is calculated using the following equations (2,5):

$$v = R_a \times T_d \times [(1.6078 \times W_i) + 1)] / p_{st}$$
$$v' = v \times p_{st} / [(W_i + 1) \times p_a]$$
$$Q_{mi} = C \times A \times \text{sqrt}(2 \times \Delta p \times v')$$
$$\Delta T = mV / k$$
$$Q_s = Q_{mi} \times v_{st} / v'$$
$$q_{thi} = Q_{mi} \times c_p \times \Delta T / [v' \times (W_i + 1)]$$

The overall coefficient of performance is given by:

$$COP = q_{thi}/(E_e + E_g)$$

AIR CONDITIONING TESTS

The method for measuring cooling performance is the same as that for heating except that the humidity ratio of the discharge air (W_o) is measured to account for latent as well as sensible heat transfer:

$$q_{tci} = q_{sci} + q_{lci}$$

$$= Q_{mi} \times [(c_p \times \Delta T) + (h_e \times (W_i - W_o))] / (v' \times (W_i + 1))$$

Other quantities measured and monitored to assure steady state operation of the heat pump:

- Indoor and outdoor relative humidity
- Indoor fan, outdoor fan, and compressor consumption
- Indoor temperature
- Outdoor temperature

HEATING VALUES OF FUELS

Propane: The higher heating value (HHV) for commercially-available propane was taken as 50150 kJ/kg (21560 Btu/lb_m) of gas (5).

Natural Gas: The higher heating value for natural gas was obtained on test days from the Canadian Gas Research Institute and averaged 37720 kJ/m^3 (1012 Btu/ft^3) of dry gas.

EXPERIMENTAL ERROR

The calculated experimental errors are given below for the various indicated measurements:

Calculated Output: Heating - 1.5% Cooling - 1.5%

Fuel Input: Propane - 3% Natural Gas - 1.5%

Electric Input - < 0.5%

These values result in calculated COP errors for the different operating modes as follows:

Heat Pump Mode - 1.5% Cooling Mode - 1.5%

Propane Gas Mode - 4.5% Natural Gas Mode - 3%

Unfortunately, our heat pump test lab has no provision for maintaining the indoor humidity level. Thus, for our cooling mode tests, the highest indoor humidity achieved in one case was 45% while the test specifications required 50%. Although the calculation error for the Cooling Mode performance was only about 1.5%, normally the cooling performance would have been higher with the correct humidity.

TEST LOOP CALIBRATION

To account for any inaccuracies in the measurement loop, a calibration constant for the loop was determined prior to each new series of tests. The calibration constant is determined in the following manner:

An electric heater installed in the measurement loop ducting just above the heat pump coil simulates the heat input to the air from the heat pump. The heater is connected to the power lines and calibrated watt-hour meters (in place of the compressor and outdoor) fan while the indoor fan power connections are not altered. The measurement loop is used to measure the heat delivered to the air from the electric heater and indoor blower using the same method as is used to do heat pump performance tests. Since any electrical energy consumed by the heater and indoor fan is delivered to the air, the energy output measured by the test loop should equal the electrical energy input. However, the output measured by the loop is usually slightly lower than the electrical input due to measurement errors and ducting heat loss.

The loop calibration constant is calculated by dividing the total electrical input to the heater and blower by the measured heat output. For subsequent performance tests, the measured heat output from the heat pump is multiplied by the constant to account for inaccuracies in the measurement loop. Calibration tests run over the project duration gave calibration constants between 1.03 and 1.05, showing that there are some small heat losses associated with the measurement loop.

THERMODYNAMIC TESTS

The steady-state, thermodynamic operating characteristics of the Kool-Fire were determined by measuring pressures and temperatures at selected locations shown in Figures 1 and E1. Pressure taps were connected to a single electronic pressure gauge via a hand-operated manifold. A Hewlett Packard computer in combination with a Hewlett Packard 3497 data-acquisition unit averaged inputs from the T-type thermocouples and the pressure transducer over a typical sampling interval of two minutes. The saturation temperatures, and superheat or subcooling for each point were calculated using the refrigerant 22 property tables (5).

WINDING TEMPERATURE MEASUREMENT

As mentioned in section 4.4 and Appendix D, the Seely method was used to measure the winding temperature of the compressor motor to determine whether there was any risk of overheating due to the reduced refrigerant charge. This method takes advantage of the fact that copper has a nearly linear coefficient of resistivity. Thus, the voltage drop of a dc current applied across a motor winding will be proportional to the winding temperature.

The device we used superimposed a small, constant dc current upon the regular ac power current applied to the compressor motor winding. First, however, a calibration curve of voltage drop versus temperature was constructed by measuring the dc voltage drop across the unpowered

motor winding at three known temperatures. Then the winding temperature was determined at several charge and operating conditions by correlating a measured voltage drop with the winding temperature on the calibration curve.

NOMENCLATURE

A - nozzle total flow area

c_p - specific heat capacity of moist air

C - nozzle coefficient

COP - coefficient of performance

E_e - electric energy consumed

E_g - gas energy consumed

h_e - latent heat of evaporation of water

k - thermopile coefficient

mV - thermopile millivolts reading

p_a - atmospheric pressure

p_{st} - standard atmospheric pressure

Δp - nozzle pressure drop

q_{lci} - latent heat transferred, cooling

q_{sci} - sensible heat transferred, cooling

q_{tci} - total heat transferred, cooling

q_{thi} - total heat transferred, heating

Q_{mi} - air flow rate

Q_s - standard air flow rate

R_a - gas constant for air

T_d - absolute temperature of discharge air

ΔT - air temperature difference

v - air specific volume

v' - corrected air specific volume

v_{st} - standard specific volume of air

W_i - humidity ratio of return air

W_o - humidity ratio of discharge air

APPENDIX B

DETERMINATION OF CHARGE

Once the charge reduction tests had been completed and before making any changes to the Kool-Fire, we had to determine the exact refrigerant charge in the system. Normally, manufacturer's specify the exact weight of refrigerant that is to be added to an installed system, but, in the Kool-Fire's case, the exact as-delivered charge was unknown. (The Kool-Fire is charged by adding refrigerant to the system until a prescribed set of operating pressures and temperatures is achieved while running in the gas mode.) Thus, the following method was developed to determine the Kool-Fire's exact as-delivered charge.

First, as much refrigerant as possible was removed from the machine and emptied into a refrigerant bottle. The refrigerant bottle was immersed in an ice-water bath and then connected to the heat pump charging port via a charging manifold. The ice-water bath kept the refrigerant bottle at about 531 kPa (77 psia), the saturation pressure corresponding to the temperature of the bottle ($2^{o}C$). Refrigerant migrated from the Kool-Fire into the bottle until the pressure in the heat pump equaled the pressure of the bottle. In this way, a total of 2.25 kg was weighed into the bottle and a small, unknown "residual" amount of refrigerant vapour was left in the machine. The residual amount of refrigerant could be determined by measuring the mass of refrigerant (.45 kg) required to return the system pressure to 531 kPa from 0kPa.

Finally, the reduced charge and as-delivered charge levels were determined. The "reduced" charge equalled the initial mass of refrigerant emptied into the bottle plus the residual quantity: 2.25 +.45 = 2.7 kg (6.0 lbs); since 1.5kg had been removed from the as-delivered machine for the charge reduction tests, the as-delivered charge was 2.7 + 1.50 = 4.2 kg (9.3 lbs).

APPENDIX C

PERFORMANCE WITH A REDUCED CHARGE

As shown by the results of the charge reduction tests (section 3.1), performed in heat pump mode at 8.3°C with the as-delivered unit, the performance of the Kool-Fire increased as the refrigerant charge was reduced. However, we wished to determine the effect the charge reduction would have on the Kool-Fire's performance at other operating conditions. Accordingly, several tests were made with 2.7 kg of refrigerant in the system i.e., 1.5kg lower than the as-delivered charge.

Table C1

PERFORMANCE OF "AS-DELIVERED" KOOL-FIRE WITH A REDUCED CHARGE AT TWO DIFFERENT AIR FLOW RATES

Measured Quantity	Units	Outdoor Temperature and Conditions 8.3°C				-8.3°C			
		Gas	Elect	Gas	Elect	Gas	Elect	Gas	Elect
Indoor Air Flow	litres/s	503	529	692	701	514	522	706	713
Temperature Rise Across Indoor Coil	°C	22.5	14.6	16.4	11.5	22.2	8.7	16.1	6.7
Electric Input	kW	3.45	3.45	3.38	3.58	3.42	2.95	3.38	3.18
Natural Gas Input	kW	11.10	-	10.70	-	11.00	-	11.10	-
Total Input	kW	14.50	3.45	14.10	3.58	14.40	2.95	14.50	3.18
Calculated Output	kW	14.10	9.40	13.80	9.70	13.80	5.50	13.80	5.80
Coefficient of Performance		.97	2.7	.98	2.7	.96	1.9	.95	1.8

Table C2a

THEMODYNAMIC OPERATING CHARACTERISTICS OF
"AS-DELIVERED" KOOL-FIRE OPERATING WITH
A REDUCED CHARGE (AIR FLOW-RATE = 700 L/s)

[Metric Units] Description (with key on Fig. 1)	Quantity	Outdoor Temperature and Conditions 8.3°C Electric	8.3°C Gas	-8.3°C Electric	-8.3°C Gas
Compressor Discharge (Point A)	P	1399	1660	1165	1663
	TS	36	43	29	43
	TA	65	84	65	85
	SH	28	41	35	42
Condenser Inlet (Point B)	P	1345	1576	1153	1582
	TS	35	41	30	41
	TA	58	82	53	82
	SH	23	41	53	82
Liquid Line (Point C)	P	655	872	447	874
	TS	9	17	-3	19
	TA	8	18	-4	18
	SH	-1	-1	-1	-1
Evaporator Traverse (Point D)	P	532	748	335	750
	TS	2	13	-11	13
	TA	2	13	-11	13
	SH	0	0	0	0
Evaporator Outlet (Point E)	P	478	666	308	666
	TS	-1	9	-14	9
	TA	-1	10	-14	11
	SH	0	1	0	2
Compressor Suction (Point F)	P	425	600	279	598
	TS	-5	5	-17	6
	TA	-4	11	-16	12
	SH	1	6	1	6

P = Absolute Pressure (kPa)
TS = Saturation Temperature (°C)
TA = Actual Temperature (°C)
SH = Degrees of Superheat (°C)

Table C2b

THEMODYNAMIC OPERATING CHARACTERISTICS OF "AS-DELIVERED" KOOL-FIRE OPERATING WITH A REDUCED CHARGE (AIR FLOW-RATE = 1500 cfm)

[Imperial Units] Description (with key on Fig. 1)	Quantity	Outdoor Temperature and Conditions 47°F Electric	47°F Gas	17°F Electric	17°F Gas
Compressor Discharge (Point A)	P	203	241	169	241
	TS	97	110	85	110
	TA	148	183	148	185
	SH	51	73	63	75
Condenser Inlet (Point B)	P	195	229	167	229
	TS	95	106	86	106
	TA	136	179	128	180
	SH	41	73	42	74
Liquid Line (Point C)	P	95	126	65	127
	TS	48	62	26	65
	TA	46	64	24	64
	SH	-2	-2	-2	-1
Evaporator Traverse (Point D)	P	77	108	49	109
	TS	36	56	12	56
	TA	36	56	12	56
	SH	0	0	0	0
Evaporator Outlet (Point E)	P	69	97	45	97
	TS	30	49	7	49
	TA	30	50	7	52
	SH	0	1	0	3
Compressor Suction (Point F)	P	62	87	40	87
	TS	23	41	2	42
	TA	25	52	4	54
	SH	2	11	2	12

P = Absolute Pressure (psia)
TS = Saturation Temperature (°F)
TA = Actual Temperature (°F)
SH = Degrees of Superheat (°F)

PERFORMANCE TESTS

Table C1 shows performance tests done at this reduced charge with the air and gas source at 8.3 and -8.3°C. (The first column of results is a repeat of the 2.7 kg case from the original charge reduction test, Table 3). Also, note that the indoor air flow rate was varied from 500L/s to 700L/s to find the effect of indoor air flow rate on performance.

There are several things to note from these results:

- The gas-source efficiency was not affected by changes in charge, outdoor temperature, or indoor air flow rate. However, the charge reduction reduced the output capacity and electrical consumption slightly.

- The air-source COP increased to 2.7 and 1.8 at 8.3°C and -8.3°C respectively due to the reduced refrigerant charge. The higher air flow rate increased the heat pump mode capacity by 5% but did not affect the overall COP because the consumption increased in the same proportion as the capacity.

THERMODYNAMIC TESTS

The thermodynamic operating characteristics at reduced charge were determined for the four tests performed at an indoor air flow of 700 L/s. These results (Table C2) revealed several things when compared to the as-delivered results performed at 500 L/s (Table 2).

The most obvious changes were the decreased high side pressures (points A, B), increased suction superheat (point F) especially in gas mode, and increased discharge superheat (point A). The slight decrease in the discharge pressure is mainly due to the increased air flow rate rather than to decreased charge. As shown in Table C1 the change in air flow rate has little or no effect on the efficiency of the heat pump.

Changes in suction superheat show the greatest variations with charge level. This is the expected result in a system that uses capillary tubes for refrigerant control. Excess refrigerant in such a system appears on the low side. Thus, the quantity of charge at any given set of operating conditions will determine whether the evaporator is partially flooded, fully flooded, or overflowing. By comparing the suction superheat values in Tables 2 and C2, it can be seen that, within the experimental accuracy of the measurements, there is no superheat at any operating condition with the as-delivered charge, but suction superheat begins to appear, especially in gas mode, with the reduced charge. As will be shown in Appendix D, this reduced charge level corresponds to the onset of a rise in winding temperature and the peak plateau in heating-mode efficiency. This is strong evidence that the compressor sump itself is being flooded by the manufacturer's charge level and that the reduced level of these tests was one that was sufficient to keep only the accumulator partially full and to keep the evaporator fully flooded (and thus, at its most efficient operating point).

That discharge superheat is unexpectedly low in Table 2, at the manufacturer's recommended charge, is a consequence of the compressor shell being flooded, as noted above. Higher, more normal values of discharge superheat are seen in Table C2 at a reduced charge. As can be seen by the improvement in efficiency shown in Table C1, this increase in discharge superheat is related to an increase in heat pump performance. This is because the presence of liquid refrigerant in the compressor shell of the as-delivered heat pump acts as a "thermal short" that robs some of the heat of compression that would have otherwise been delivered to the indoor coil. This liquid refrigerant, which is boiled within the compressor shell itself, must be pumped around the system by the compressor but it does not ever deliver heat to the conditioned space. In this way, the heat pump consumption is higher and its output is lower than would otherwise be the case and this translates into the lower COPs obvserved with the as-delivered unit.

APPENDIX D

VERIFICATION OF AIR-CONDITIONING PERFORMANCE AND RELIABILITY WITH A REDUCED CHARGE

During the project we suggested to the manufacturer that an increase in performance might be obtained if he was to reduce the charge that he normally puts in his machines. However, he was concerned that a refrigerant charge reduction would:

- degrade the cooling mode performance,
- reduce compressor-motor cooling, and
- reduce compressor-motor lubrication.

As an extension to the project, we investigated the effect of refrigerant charge on these three aspects.

PERFORMANCE RESULTS

Addressing the first concern, the Kool-Fire's cooling and air-source heating performance were tested while varying the refrigerant charge. All performance tests were conducted in accordance with ARI Standard 240-81 (Appendix A). Heating tests were run at the 8.3°C rating conditions; cooling tests were run at 35°C outdoor temperature, 26.7°C indoor temperature rating conditions. Cooling tests also require 50% relative humidity indoors. Unfortunately, since our test facility was designed principally for heating mode tests, there was no control over the indoor humidity. The highest relative humidity we were able to achieve was about 45% in one case. The Kool-Fire cooling performance would have been somewhat higher had we been able to provide 50% relative humidity as required by the standard.

Figure D1 and Table D1 show the response of the heating and cooling mode performance to variations in charge. As was expected from previous tests (Appendix C), the optimum heating performance peaks at 2.25 kg (5.0 lbs). However, the cooling mode performance peaks at a charge of 2.95 kg

Table D1

KOOL-FIRE PERFORMANCE AND WINDING TEMPERATURE
VARIATION WITH REFRIGERANT CHARGE

Air-Source Heating Mode at High Temperature Rating Conditions: Indoor 21.1°C, Outdoor 8.3°C										
Refrigerant Charge (kg)	2.15	2.27	2.38	2.50	2.50	2.72	2.95	3.86	4.08	4.31
Indoor Air Flow (L/s)	715	697	717	701	716	708	708	699	700	697
Electric Input (kW)	3.58	3.56	3.62	3.64	3.67	3.69	3.73	3.74	3.72	3.71
Calculated Output (kW)	9.61	9.63	9.63	9.64	9.65	9.61	9.53	9.14	8.94	8.81
Coefficient of Performance	2.68	2.7	2.69	2.65	2.63	2.61	2.55	2.44	2.4	2.37
Winding Temperature (C)	17.7	17.3	9.1	2.6	-2					

Cooling Mode at Standard Rating Conditions: Indoor 26.7°C, Outdoor 35°C								
Refrigerant Charge (kg)	2.0	2.25	2.5	2.7	2.95	3.2	3.6	4.1
Indoor Air Flow (L/s)	708	710	710	714	712	702	704	702
Electric Input (kW)	4.04	4.21	4.32	4.44	4.56	4.66	4.64	4.66
Calculated Output (kW)	6.81	6.93	7.48	7.85	8.46	8.24	7.57	7.38
Coefficient of Performance	1.55	1.65	1.73	1.77	1.86	1.77	1.63	1.58
Winding Temp. (°C)	59	51	49	48	43	35	11	11

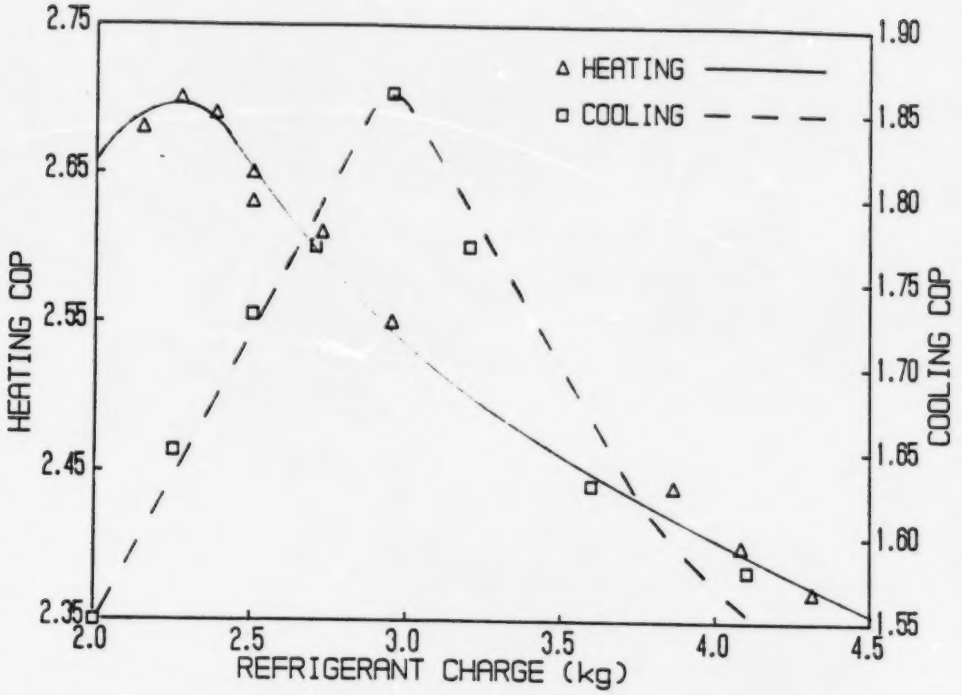


Figure D1 Effect of charge on efficiency for the "as-delivered" Kool-Fire

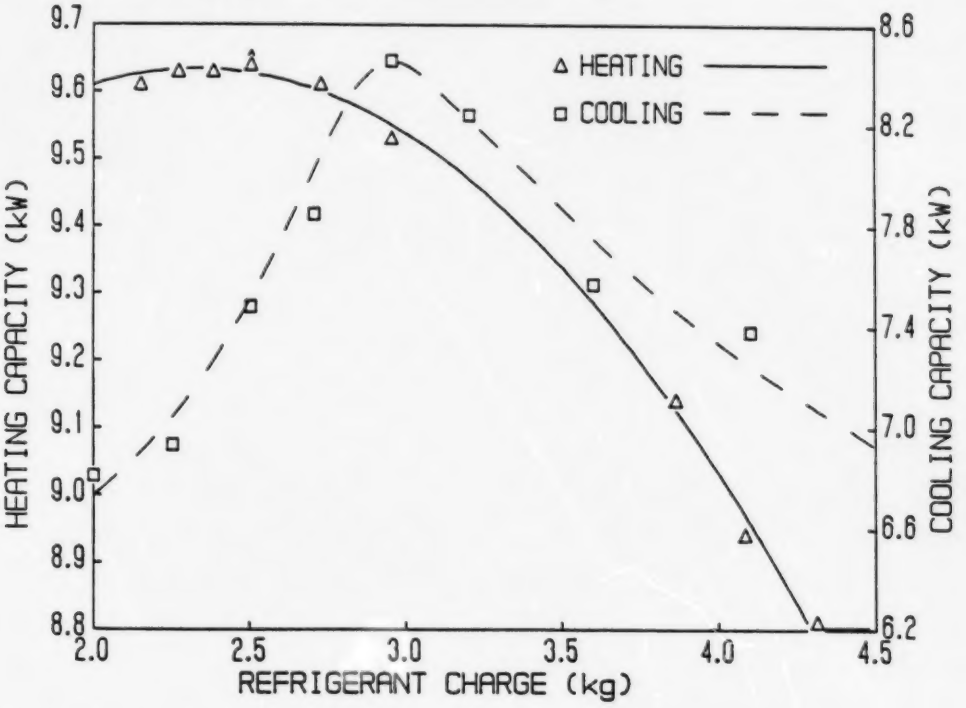


Figure D2 Effect of charge on capacity for the "as-delivered" Kool-Fire

(6.5 lbs) with a COP of 1.9 (EER of 6.4). A look at Figure D1 also shows that a compromise between the best heating and cooling performances would be achieved with about 2.7 kg (6 lbs) of refrigerant.

Both performance and thermodynamic data showed that, contrary to the manufacturer's concerns, even the cooling performance was better with a refrigerant charge lower than the Kool-Fire's as-delivered charge. At lower charge levels, more heat was absorbed by the indoor coil, superheat increased at the evaporator outlet, and subcooling decreased at the condenser outlet. Also, the lower operating pressures reduced the compressor consumption while allowing more energy to be absorbed by the evaporator per mass of refrigerant pumped. These results reflect the effect of reduced refrigerant charge on air-source heating mode shown in Appendix C.

Even though a peak cooling COP of 1.9 was achieved, this value is still lower than most residential heat pumps. (Published values for EER's are normally around 9, COP around 2.6.) As previously mentioned, the low indoor humidity levels were partly responsible for the poor cooling performance, since a good part of an air conditioner's performance is due to latent heat (in the form of moisture) removed from the indoor air. We would expect the Kool-Fire's cooling performance to be somewhat higher at the 50% humidity prescribed by the standard. However, even if the Kool-Fire performance was given a generous credit of 20%, the best cooling COP still would be low at about 2.2.

This low cooling mode performance might partially be blamed on the manufacturer's use of a single set of capillary tubes as the expansion device for both the heating and cooling operation. As mentioned in section 3.2, most heat pumps have two expansion devices - one sized for cooling and one sized for heating. The Kool-Fire capillary tubes, on the other hand, must handle the gas mode operation in addition to the normal heating and cooling functions.

The manufacturer likely chose to use only a single set of capillary tubes to keep the reliability of his product high since expansion valves and check valves (required with systems using two expansion devices) can stick. According to the manufacturer, the capillary tubes have been sized for the gas-source heating mode. This is probably the best practice for sizing the Kool-Fire's single set of capillary tubes since the gas-source expansion device requirements lie between air-source heating and air-conditioning requirements. However, the Kool-Fire should be equipped with separate expansion devices for heating and cooling for best overall performance.

For the cooling mode, capillary tubes are a good choice since cooling takes place over a relatively narrow range of operating conditions. This is standard practice with most commercially-available brands of heat pumps. However, for the heating mode, the expansion device must operate effectively over a much wider range of conditions, especially in a cold climate such as ours. Thermostatic expansion valves are able to handle a wide range of conditions better than capillary tubes.

COMPRESSOR WINDING TEMPERATURE MEASUREMENT

The manufacturer was also concerned that a reduced charge would overheat the compressor motor. Since the motor windings are cooled by refrigerant at the suction conditions, a reduced charge would reduce the refrigerant mass flow over the windings and, therefore, reduce the heat removed from the windings. Most compressor motors are protected from overheating by a built-in thermal switch and a surface-mounted thermostat placed on the discharge line. (The compressor manufacturer recommends the discharge line thermostat be set to turn off the compressor at 113°C (235°F)). However, consistently high operating temperatures will accelerate the deterioration of lubricants, motor-winding insulation, and refrigerant.

For any heat pump compressor, the worst cooling would occur in the air-conditioning mode at a low refrigerant charge or at very low-temperature

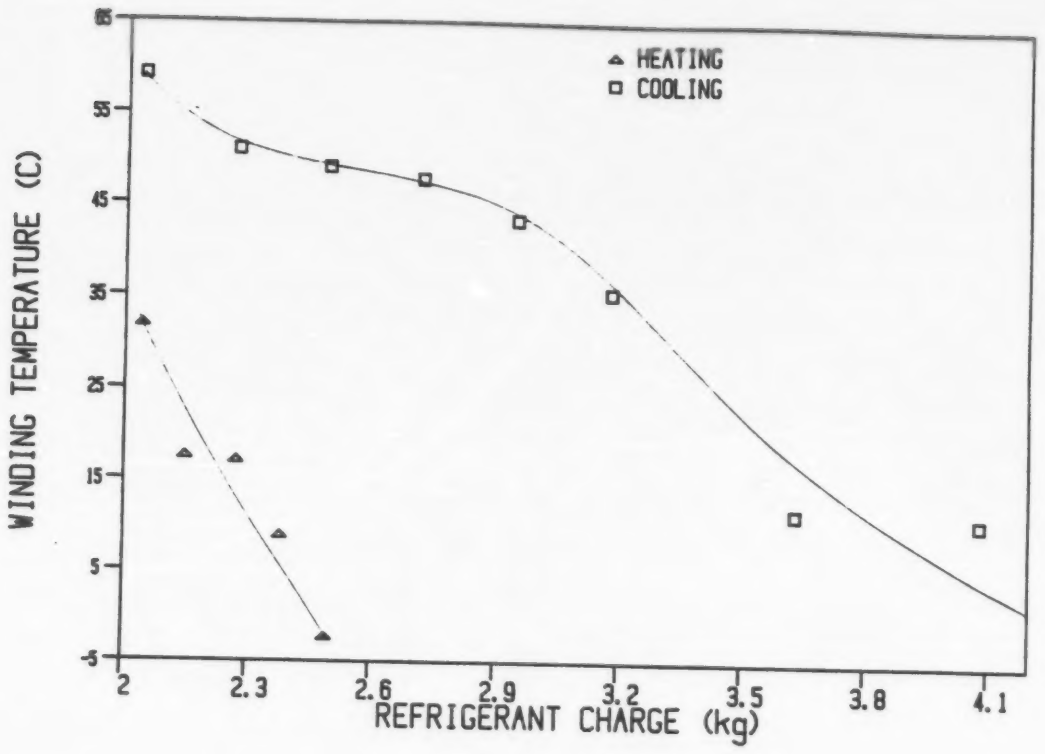


Figure D3 Effect of charge on winding temperature for improved Kool-Fire in air-source heating and cooling modes.

heating. An examination of the air-conditioning thermodynamic data (Table D2) shows the actual discharge gas temperature (as measured by an insertion thermocouple) to be 123°C at the lowest charge (2kg). However, a surface-mounted thermocouple or cut-off thermostat at this point would register a temperature of about 93°C, well below the cut-off temperature (113°C) mentioned above.

A more direct measurement of the compressor-motor winding temperature was made using the Seely Method as described in Appendix A. The variation of winding temperature with charge is shown in Table D1 and Figure D3 for both the air-source cooling and heating modes. As expected, the cooling mode exposed the windings to the highest temperatures. However, even the highest winding temperature (59°C), recorded at the lowest charge

conditions (2 kg), was within accepted limits.

COMPRESSOR LUBRICATION

There was some concern that a reduced charge would decrease compressor lubrication. Normally, the lubricant is carried out of the compressor by the refrigerant since the oil and refrigerant are completely miscible with each other. However, this does not present a problem providing the speed and density of the refrigerant gas in the suction line are high enough to return the migrating oil to the compressor.

In a refrigeration system, the worst conditions for oil return occur at cold evaporating temperatures and at low charge conditions where the flow of refrigerant is at its lowest. However, by specifying the correct suction line diameter for a given system, which determines refrigerant speed, satisfactory oil return will be maintained at all operating conditions. For the 5/8" suction line used with the Kool-Fire, the lowest calculated refrigerant flow rate in our tests was 2.5 times the highest rate required (7).

Another decrease in lubrication can take place if liquid refrigerant finds its way back to the compressor from the accumulator. Normally, the accumulator can accommodate most of the heat pump's refrigerant charge. However in an overcharged system, liquid refrigerant can overflow the accumulator, run into the compressor and dilute the refrigerant oil. The mechanical parts of the compressor then may be damaged. Thus, by adding too much refrigerant to the "as-delivered" machine, the manufacturer may actually be causing decreased compressor lubrication, the very thing he wanted to prevent. The extremely low winding temperatures observed as charge levels approach the recommended levels are strong evidence of liquid flood-back to the compressor sump under these conditions.

APPENDIX E

HARDWARE MODIFICATION DETAILS

As a final step in our optimization of the Kool-Fire, the system's refrigerant circuiting was physically altered to obtain more information about the Kool-Fire's operating characteristics and the effects of some of the proposed improvements. The reader should refer to Figure E1 while following the description of the changes below. The effects of these changes are described elsewhere in the report.

INDOOR COIL

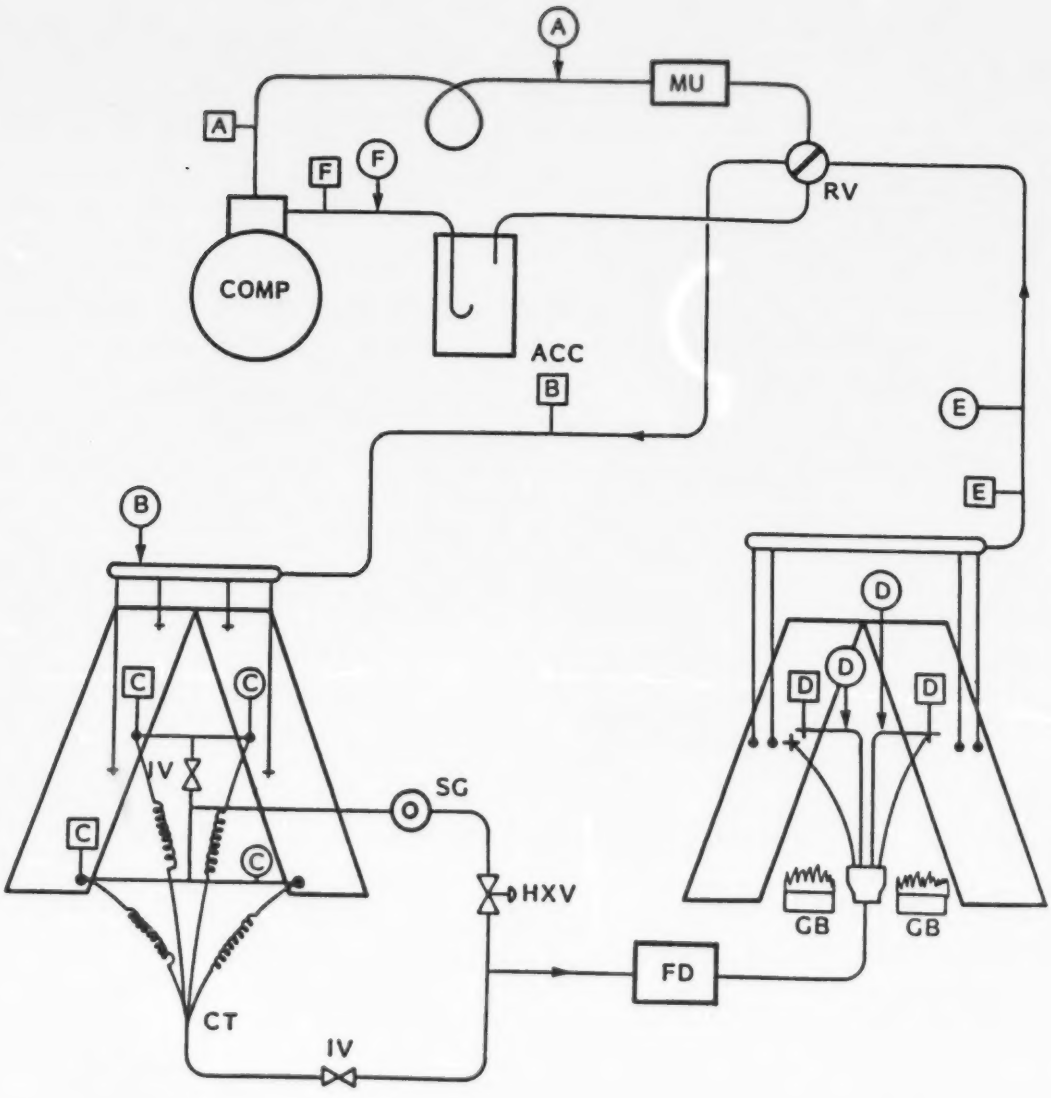
The change to the piping at the indoor coil was by far the most complex of our modifications. A completely new piping network had to be designed that would allow us to choose between two different expansion devices and to more accurately measure coil operating parameters. Accordingly, the following changes were made:

- An insertion thermocouple was installed on the inlet header to the coil. This gave us better measurement of the temperature of the superheated refrigerant as it entered the coil.

- A hand-actuated expansion valve and a sight glass were added in parallel to the existing capillary tubes. By correctly operating an isolating valve and the expansion valve, the experimenter could choose to control the refrigerant via the capillary tubes, the expansion valve, or a combination of the two.

- Two pressure ports were added at the outlets from the coil. These allowed us to accurately measure the pressure of the refrigerant right at the coil outlet instead of in the liquid line as for the original tests.

- In addition, a valve installed in the piping allowed us to isolate the



ACC	*Accumulator*
COMP	*Compressor*
CT	*Capillary Tube*
DI	*Distributor*
FD	*Filter-Drier*
HXV	*Hand Expansion Valve*
GB	*Gas Burners*
IV	*Isolation Valve*
MU	*Muffler*
RV	*Reversing Valve*
SG	*Sight Glass*

A	*Compressor Discharge*
B	*Condenser Inlet*
C	*Condenser Outlet*
D	*Evaporator Inlet*
E	*Evaporator Outlet*
F	*Compressor Suction*
□	*Pressure Taps*
○	*Surface Thermocouples*
○	*Insertion Thermocouples*

Figure E1 Schematic of the improved Kool-Fire. Measurement locations added in the laboratory are given by the single letter designations. The reversing valve is shown in heating mode position.

top pair of coil circuits from the bottom pair. By closing the valve, the pressure of the top circuits could be measured independently of the bottom circuits. In this way we could determine whether the flow through the top and bottom pairs of coil circuits was unbalanced by watching for a difference between the two measured outlet pressures.

OUTDOOR COIL

At the outdoor coil, no circuiting changes were required, but we wished to improve the measurement of the state of the refrigerant at the inlet and outlet of the coil. Accordingly, an insertion-thermocouple/pressure-tap pair was installed on both the left and right inlets to the coil and an insertion thermocouple was added to the outlet header. The pressure tap on the outlet was left unchanged.

The installation of the two inlet measurement points was a little tricky. Since the distributor tubes were soldered directly into the ends of the coil, we could not de-solder the tubes without damaging the coil. This problem was circumvented by carefully cutting two of the distributor tubes close to the coil and splicing fittings into the tubes. Each fitting contained an insertion-thermocouple/pressure-tap pair.

FILTER-DRIER

In order to demonstrate the effect of reducing the suction line pressure drop, the suction line filter-drier was removed and a filter-drier was added to the liquid line. In most heat pumps, the liquid line is the preferred location for the filter-drier. (The refrigerant is more easily filtered since it's in its liquid phase at this point in the circuiting, and the pressure drop from the filter is negligible compared to the expansion device pressure drop.) Of course, in a heat pump, a bi-directional filter must be used in the liquid line since the refrigerant flow is reversed in going from the heating to the cooling mode.

The implementation of this change was fairly straightforward. Due to the proximity of the suction-line filter-drier to the reversing valve, a good deal of care had to be taken not to overheat the reversing valve while removing the filter. The replacement filter was easily installed by cutting into the liquid line and connecting the filter in place with mechanical (rather than soldered) couplings.

COMPRESSOR

Insertion thermocouples were installed at the compressor suction and discharge in order to better measure the temperature of the superheated refrigerant at these locations. The tubing supplied with the Kool-Fire would have been too small to accommodate insertion thermocouples without impeding refrigerant flow. Accordingly, the two thermocouples were installed in two larger diameter pipes spliced into the suction and discharge lines.

One undesirable consequence of all these changes was the unavoidable increase in the number of soldered joints in the system. Any joint is a potential leak site and, in fact, a good deal of time was spent finding and repairing small leaks that did develop. Since our data was greatly dependent upon knowing the exact refrigerant charge, a number of our tests had to be redone to verify results.

APPENDIX F

CALCULATION OF YEARLY GAS CONSUMPTION FOR THE IMPROVED KOOLFIRE

As mentioned in section 5, it is possible that the gas consumption of a Kool-Fire heat pump with gas defrost may be so low as to make connection to natural gas supply lines uneconomical. In this case, a home's entire yearly heating needs could be met with a Kool-Fire connected to a propane tank. Accordingly, an approximate calculation of the Kool-Fire's yearly gas or propane consumption was made.

First, it is necessary to calculate a typical balance point temperature for this system. The balance point occurs at the temperature where the heating load of the house equals the capacity of the air-source

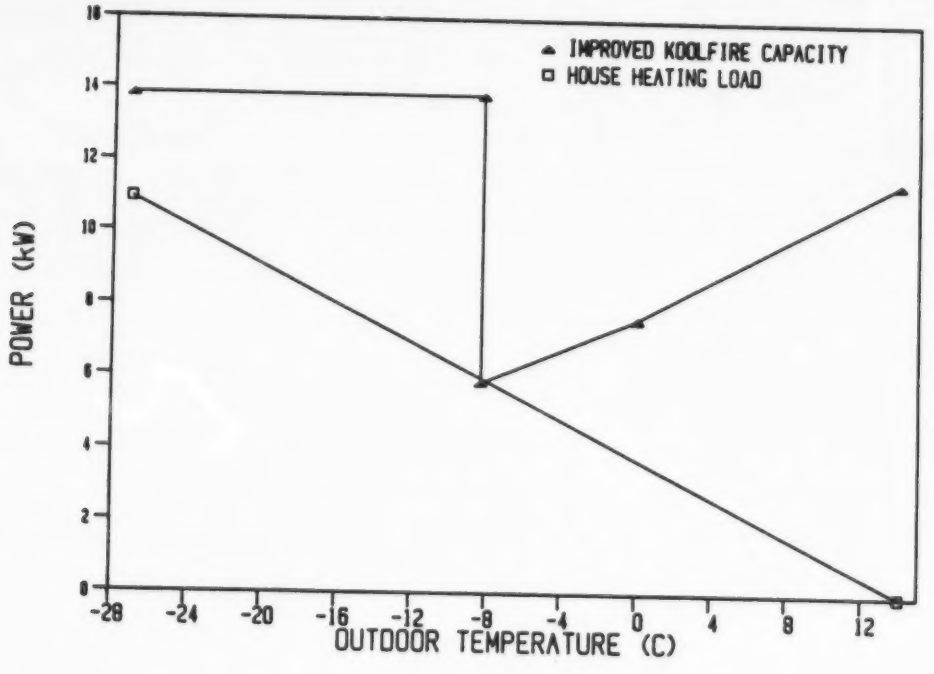


Figure F1 Balance point calculation for the improved Kool-Fire operating in a typical house. The intersection of the two lines shows the balance point to be about -8°C.

operation. Figure F1 shows the capacity curve of the improved Kool-Fire and the heating load of a typical home that would use a 2.5 ton heat pump. (The "typical" house used in this report consumes 20,000 kWh per year for heating in the Toronto area and has a reference temperature of 14°C.) The intersection of the two curves shows the balance point temperature to be about -8°C.

Next, a bin method is used (Table F1) to calculate the heating load that must be supplied by the propane fuel for our typical house. The heating season degree hours for Toronto are given in the second column of Table F1. These values use a 14°C heating base and 2°C temperature bins. The third column shows the heating requirement for each temperature bin, calculated by dividing the total yearly degree-hours (DH_t) into the product of the bin degree-hours (DH_i) and the total yearly heating load for our typical residence (20,000 kWh/year).

$$HR_i = DH_i \times 20000 / DH_t.$$

Column 5 shows the fraction of the heating load (f_i) that can be served by the air-source operation. For temperatures above 8°C, all of the heating is done by the air-source. However, at frosting temperatures (between 8°C and -8°C) and below the balance point (-8°C) some gas-source heating is required. For the frosting range, the heat pump fraction was taken as 0.95 since a calculation showed that about 5% of the heating was done by the gas-mode during defrosting. The fractions below the balance point were calculated by dividing the heat pump capacity by the house load at the bin temperature. Furthermore, it was assumed that the air-source would not run below -16°C since the COP approaches 1.0 and no advantage is to be gained by using the air-source.

Column 5 then shows the amount of heating energy supplied by the air-source heating mode (HSa_i) arrived at by multiplying the heating requirement by the air-source fraction:

$$HSa_i = f_i \times HR_i$$

The last column shows the amount of energy supplied by the gas-mode in each bin calculated as the portion of the bin load not supplied by the air-source :

$$HSg_i = HR_i - HSa_i$$

The bottom line shows the total amount of heat that must be supplied in the gas-mode over the heating season: 2837.5 kWh.

From this value the yearly propane consumption can be calculated. Our experimental data shows that the Kool-Fire consumes 32 units of electric energy for every 100 units of gas energy consumed. Since the higher heating value of commercial propane is 13.93 kWh/kg and the average gas mode efficiency is 0.95, the capacity of the Kool-Fire per kilogram of propane burned is:

$$13.93 \times (1+.32) \times 0.95 = 17.47 \text{ kWh/kg (27100 Btu/lb)}$$

Thus, the total amount of propane required over the heating season is calculated by dividing the capacity of the gas mode per kilogram of propane into the gas mode heating load:

$$2837.5 \text{ kWh} / 17.47 \text{ kWh/kg} = 162 \text{ kg (358 lb)}.$$

To give the reader some idea, this weight of propane is equivalent to 318 litres which, at $.32 per litre, would cost about $102 per year. This much propane could be stored in one 360-litre, "pot-belly" tank (1.2 m high by 1 m in diameter). In fact, many rural customers already use larger 1000 and 2000 litre tanks to meet their space and water heating needs.

Table F1

CALCULATION OF YEARLY GAS CONSUMPTION
FOR THE IMPROVED KOOL-FIRE

Bin i	Bin Centre (°C)	Degree Hours DH_i (°C-h)	Heating Requirement HR_i (kWh)	Heat Pump Fraction f_i	Heat Pump Mode Heat Supplied Ea_i (kWh)	Gas Mode Heat Supplied Eg_i (kWh)
1	13	497	133.1	1.000	133.1	
2	11	1434	384.1	1.000	384.1	
3	9	2290	613.4	1.000	613.4	
4	7	3157	845.6	0.950	803.3	42.3
5	5	4122	1104.1	0.950	1048.9	55.2
6	3	5918	1585.2	0.950	1505.9	79.3
7	1	8671	2322.6	0.950	2206.5	116.1
8	-1	8580	2298.2	0.950	2183.3	114.9
9	-3	7497	2008.1	0.950	1907.7	100.4
10	-5	6498	1740.5	0.950	1653.5	87.0
11	-7	5796	1552.5	0.950	1474.9	77.6
12	-9	5221	1398.5	0.925	1293.6	104.9
13	-11	4475	1198.7	0.779	933.8	264.9
14	-13	3618	969.1	0.654	633.8	335.3
15	-15	2639	706.9	0.547	386.7	320.2
16	-17	1953	523.1			523.1
17	-19	1221	327.1			327.1
18	-21	665	178.1			178.1
19	-23	296	79.2			79.2
20	-25	78	20.9			20.9
21	-27	41	11.0			11.0
TOTALS:		74667	20000.0		17162.5	2837.5

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